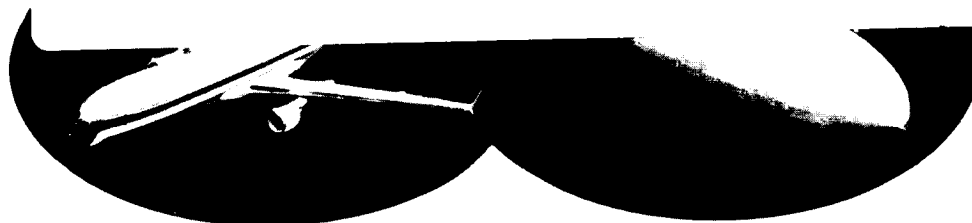


**FAA TECHNICAL CENTER**  
Atlantic City International Airport  
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# **Proceedings Of The Ninth DoD/NASA/FAA Conference On Fibrous Composites In Structural Design**

Lake Tahoe, Nevada  
November 4-7, 1991



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**FAA TECHNICAL CENTER**  
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# **Ninth DoD/NASA/FAA Conference on Fibrous Composites in Structural Design**

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Lake Tahoe, Nevada on November 4-7, 1991

September, 1992

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## PREFACE

The Ninth DoD/NASA/FAA Conference on Fibrous Composites in Structural Design is one of a series of conferences jointly sponsored by the Federal Aviation Administration, the National Aeronautics and Space Administration, the U.S. Air Force, the U.S. Army, and the U.S. Navy (Department of Defense). The purpose of this series of conferences is to convene periodically key government and industry research and design engineers to present and discuss the status, problems, and requirements in the technical disciplines related to the design of composite structures. This series of conferences provides a forum for the scientific community to exchange composite structures design and technology.

The Ninth DoD/NASA/FAA Conference on Fibrous Composites in Structural Design was hosted by the Federal Aviation Administration and held at Lake Tahoe, Nevada during November 4-7, 1991. The conference offered 91 presentations by senior managers and experts in the field of composite structures, organized into a total of 11 sessions. These included: one overview session on perspectives in composites; seven discipline sessions in applications (two sessions); innovative design/manufacturing (one session); methodology (two sessions); reliability (one session); damage tolerance (one session); and two focused sessions on thick structures and space structures. The conference also hosted the second industry briefing on the NASA Advanced Composites Technology (ACT) program. This publication contains (in three volumes) the technical material presented in these sessions.

*Certain materials are identified in this publication in order to specify adequately which materials were used in the structural design or research efforts. In no case does such identification imply recommendation or endorsement of a product by FAA, NASA, or DoD, nor does it imply that the materials are necessarily the only ones or the best ones available for the purpose. In many cases, equivalent materials are available and would probably produce equivalent results.*

The Conference Organizers would like to take this opportunity to thank all the authors and presenters for their outstanding contributions to the conference technical program, as well as the conference attendees whose contributions to the conference discussions helped to make the conference a successful technology exchange forum for current composite structural design issues.

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Wright Research and Development Center  
Army Aviation Applied Technology Directorate

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**SESSION VII**  
**INNOVATIVE DESIGN/MANUFACTURING**

*Volume III*

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# PROCESS AND CONTROL SYSTEMS FOR COMPOSITES MANUFACTURING

by

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## Summary

A precise control of composite material processing would not only improve part quality, but also directly reduce the overall manufacturing cost. The development and incorporation of sensors will help to generate real-time information for material processing relationships and equipment characteristics. In the present work, the thermocouple, pressure transducer and dielectrometer technologies were investigated. The monitoring sensors were integrated with the computerized control system in three non-autoclave fabrication techniques: hot-press, self contained tool (self heating and pressurizing) and pressure vessel. The sensors were implemented in the parts and tools.

## Introduction

Advanced composites, in general, offer enhanced material stiffness, damage tolerance and environmental durability. There have been considerable advances in the design and utilization of the advanced composites in aircraft and aerospace structures. However, the fabrication processes are still conducted on the conventional basis of monitoring the autoclave or vessel environment without a true in-process closed-loop control system on the individual part. Polymeric composites are typically processed to a predetermined time-temperature-pressure profile for a given resin system. These profiles are varied from material batches or environmental exposures. The processing cycles should be determined by measuring some parameters which are truly indicative of cure or consolidation progression [1,2].

In general, the curing profiles monitored inside the autoclave are different from the temperature and

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pressure actually applied on the parts. The actual amount of heat and pressure seen in the resin matrix is a function of material type and fiber layup. They are critical to the integration and compaction of laminates. Therefore, it is important that the actual pressure applied on the resin matrix during the appropriate thermal processing window be monitored and used to control the fabrication. Thermocouples and pressure transducers are the most popular sensors to perform the data collection and process monitoring. The dielectric response of the thermoset resins has also been used to monitor the progression of composite cure [3]. Some of the commercially available micro-dielectric sensors have unique fixed geometries and integral systems [4]. They can reflect the corresponding viscosity data of resins at various electrical frequencies during a thermal process.

The in-situ data of the heat and pressure transfer between tool and part is essential to a qualitative process control. Placing sensors in parts, tools and equipment will enhance the processing technology. An in-process monitor and closed-loop control system can be developed by integrating the sensors to the computerized control units. Such a system would improve the quality of composite parts and potentially reduce the manufacturing cost.

Development, utilization and interpretation of sensor response is a key element in validating materials and developing a scientific processing database. The sensors must be fast in response, accurate over the processing range and cost-effective in the production environment. In the present work, the newly developed thin foil thermocouple, twin-interferometry pressure transducer and micro-dielectric sensors have been investigated and implemented with computer control systems. A self-contained tool and an integrally heated tool were developed for composite fabrication, which have been demonstrated with non-autoclave processing techniques.

### **Thermocouples**

Most thermocouples are constructed of metallic wires with known characteristics. Production-hardened thermocouples are very rugged sensors and are often the least expensive method of temperature measurement. Due to their low mass, thermocouples generally have very rapid response to changes in temperature. The latest development in thermocouple design has been in the area of the thin foil types, which are made from two standard thermocouple materials. The foils are 0.002 inches thick. Two foil leads were first welded in a serpentine junction pattern, and then laminated with Kapton film to be rugged and easy to handle. As shown in Fig.1, the 6-inch long lead offers the distinct advantage of embedding the thermocouple between composite laminate plies with negligible effect. It provides an effective tool to monitor the temperature profile in the composite part during processing.

### **Pressure Transducers**

There are many different designs to bring pressure to the transducer. The common idea is to make a

mechanical measurement of the displacement of a diaphragm. The conventional pressure transducers are designed by using strain gage techniques. A Wheatstone bridge wiring is bonded on or embedded in the metallic diaphragm to measure deflections. With a proper design of the measurement system, the sensors are also able to compensate for thermal effects in the operating range. Some applications require a flush mount of the diaphragm onto the tooling surface and others require a hose connected to a pressure port. For the production environment, ruggedness is a major factor to be considered. The total size of the transducer package may become a driving issue in some applications.

Unlike the conventional design, a twin-interferometry system and the laser-optic readout scheme were developed at Massachusetts Institute of Technology (MIT) under a Lockheed contract [5,6]. This Twin-interferometry system uses two parallel optical fibers normal to the diaphragm. The laser beams are transmitted through the fibers onto the diaphragm surface. By counting the interference fringes caused by the reflected lights from the diaphragm, the magnitude and direction of the diaphragm deflections can be determined. As shown in Fig.2, the prototype includes lasers, couplers, optical fibers, a stainless steel diaphragm and electrical box. This system has been demonstrated and proved in numerous laboratory tests. Several critical technical issues have been identified on these sensors for future development.

### **Dielectric Sensors Technology**

During the cure process, changes of polymeric resin properties are reflected in their dipole alignment and ion movement. By measuring their dielectric values, the degree of reaction and resin viscosity can then be characterized. Dielectric response of polymeric composites has been used to monitor the progression of thermoset composite cure [3,4]. It has contributed significantly toward identification of specific material or process deficiencies. This method produces consistent and reproducible data. The newly developed micro-dielectric sensors by Micromet are rugged and production ready. It is an effective process monitoring technique.

### **Non-Autoclave Fabrication Techniques**

The conventional fabrication method for advanced composites is to utilize an autoclave, which provides the required heat and pressure for curing or consolidation. However, the autoclave process has many technical limitations and high operational cost. The primary problem is that the heat in the autoclave is not uniformly distributed and difficult to predict flow. The uneven cure or consolidation of composite components can happen due to the complexity and varieties of the geometries of parts and tools. In order to maintain uniform heating, additional setup and ducting in the autoclave may be necessary. Thus, the batching task of production parts and the nesting placement in the autoclave becomes very important and time consuming. Besides, it is not feasible in equipment cost and capacity scale-up to keep numerous autoclaves available and adaptable for the changing production requirements and schedules. Alternative non-autoclave fabrication techniques have been developed for such needs and feature improved part quality and reduced



manufacturing cost.

In order to achieve a uniform and complete heating on a complex composite part, a self-heated tool should be the most efficient technique. The tool is heated by independent power units at different heating zones, which are designed upon the thicknesses and dimensions of the part. A more advanced tool, such as a self-contained tool, includes additional pressure and cooling systems. The self-contained tools have been developed and implemented for fabricating graphite/epoxy composite parts in earlier programs [7,8]. The parts were cured at 350 °F under a pressure of 100 psi. Because of the higher temperature (~600 °F) and pressure (200 psi) necessary to consolidate graphite/PEEK thermoplastic composites, a self-contained tool is not practical. The alternative is to heat the part using a powerful integrally heated tool and pressurizing it inside a pressure vessel. Recently, a composite reinforced ceramic integrally heated tool was developed for the high-temperature thermoplastic composite consolidation [9]. The tool was instrumented with a power control system.

### **In-Process Closed-Loop Control System**

A computerized data acquisition and control system has been developed by Applied Polymer Technology (APT) Inc. Named the Composite Automated Process System (CAPS), it is designed specifically for composite fabrication. CAPS performs an intelligent management of the predefined process parameters: temperature, pressure and vacuum. It will adjust the processing cycle according to the real-time sensor signals. As shown in Fig. 3, each control window should be signaled by the sensors during the process cycle. In other words, the degree of cure or consolidation of the composite part is monitored by a dynamic control system with the sensor placement.

In the present work, the computer was integrated with sensors, microprocessors and power controller to establish an in-process closed-loop control system. All sensing data was recorded by the CAPS 210 computer during the process cycle. CAPS will print a summary throughout the process as well as a quality control data report at the end of each run. The Barber-Colman 560 series microprocessors were set up to control the setpoints using the current proportioning technique, instead of the conventional time proportioning technique. Each setpoint controlled one heating power unit. Thermocouples measured temperatures in the part and at the heaters. Vacuum and pressure were monitored by transducers mounted on the tool or the pressure vessel. An instrumented tool can detect the thermal and mechanical changes of tool and part. Sensor placement consider bag, tool and part positions, which become critical decisions in terms of monitoring.

### **Demonstrations**

Three in-process closed-loop control systems have been developed in this program. The incorporation and

placement of sensors in the parts and tooling systems provides an advanced processing technique.

#### Case 1: Hot press and dielectrometry system

As illustrated in Fig.4, a Tetrahedron hot press was used to fabricate graphite/epoxy (Hercules Magnamite graphite prepreg tape AS4/2220-3) flat panels. The press has a 24 inches by 24 inches platen with a programmable control system. It was integrated with an IBM personal computer, Micromet Eumetric system and the microdielectrometers. The sensors were embedded between laminate plies, as seen in Fig.5. Because of the conductive property of graphite fiber, a piece of glass fabric must be placed on the top of the sensor to prevent a short circuit. A low conductivity integrated circuit dielectric sensor was used for measurement at five different frequencies (1, 10, 100, 1000, 10000 Hz). The curing cycle was to heat from ambient at 5 °F/minute to 350 °F and hold for 120 minutes and cool down under 85 psi pressure. A Micromet developed critical point control software was modified and implemented as the monitor/control program.

#### Case 2: Self-contained tool and twin-interferometry pressure transducer system

A self-contained tool was used for graphite/epoxy (Hercules Magnamite graphite prepreg tape AS4/2220-3) flat panel fabrication. As shown in Fig.6, inside the bolt tightened steel tool, there is an expandable silicon rubber bag, a printed circuit heater, cover caul plates, composite laminates and insulation. A Tayco foil type thermocouple was bonded on the heater for closed-loop control. The heater was connected to a power unit and the Barber-Colman controllers. The MIT developed twin-interferometry pressure transducer and a commercially available Kulite semiconductor sensor were threaded into the tool and flushed with the tooling surface, as seen in Fig.7. The pressurized air bag applied pressure on the cauls and laminates. Both sensors measured the actual applied pressure during the heating process. In Fig.8, the CAPS system was interfaced with a power cart and printed circuit heater. Two Barber-Colman micro-processors were used for setpoint control on the heater. Pressurized air was manually controlled and safe guarded by a regulator.

#### Case 3: Integrally heated tool and thermocouples in a pressure vessel

Pressure vessel processing, as shown in Fig.9, is a proven low cost, time saving method of forming and consolidating thermoplastic composite parts [9]. This is accomplished through the use of an integrally heated tool that heats the part up to processing temperature within an unheated pressurized vessel that supplies pressure to the part. The pressure vessel described in this work is 11 ft. long and 6 ft. diameter. The vessel is capable of providing 200 psi pressure. As illustrated in Fig.10, the CAPS was interfaced with a power cart and the integrally heated tool. The integrally heated tool was made of graphite reinforced ceramic matrix from Comtool Technology [8]. It displayed low watt density requirements to achieve ramp cycles of less than 30 minutes to 750 °F. This tool demonstrated improved mechanical properties, reduced

specific heat and lowered coefficient of thermal expansion. The pressure vessel is supplied with electrical ports for the tool heaters and zone controllers. It is also equipped with numerous vacuum and thermocouple ports such that part temperatures can be easily monitored and controlled in specific locations of the part. The heating power of the tool was closed-loop controlled by the Barber-Colman microprocessors and CAPS controller. This processing system was demonstrated by fabricating a contoured shape thermoplastic composite (ICI APC-2, AS4/PEEK) part, as shown in Fig.11.

## **Conclusion**

In this program, the following technologies have been investigated:

- . Thermocouple, pressure sensor and dielectrometer
- . Advanced tooling technology
- . Integration of sensors and tooling
- . Non-autoclave fabrication techniques
- . In-process closed-loop monitor/control systems

The in-process closed-loop monitor/control systems have been demonstrated in three different fabrication techniques. Thermoset or thermoplastic composites were chosen in each demonstration, based upon the manufacturing cost effectiveness. The demonstrated processes require a fraction of the time required with conventional autoclaves and resulted in good quality parts at a lower cost. In the pressure vessel application, because of no elevated temperatures, the plumbing of vacuum ports and thermocouple outlets is simplified for rapid and easy hook-up. The bagging and sealant tapes no longer have the high temperature requirements of the autoclave. Therefore, they are less expensive, easier to work with and require roughly 75% less time to bag a part.

Those accomplishments were conducted only on the concept developmental level. Many details in optimizing and controlling the fabrication cycles still need continued study. The time-temperature-pressure processing profile should be controlled on a logical basis [10]. The goals are to establish consistent part quality and reduce the manufacturing cost.

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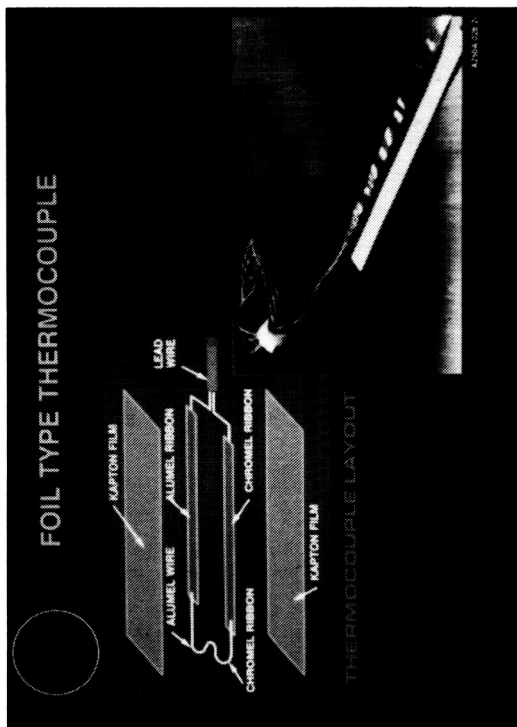


Fig. 1 Thin Foil Type Thermocouple

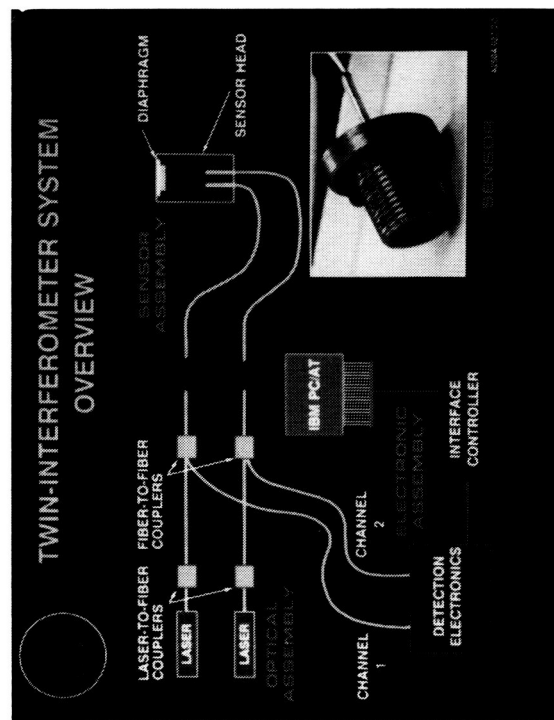


Fig. 2 Twin-Interferometry System Overview

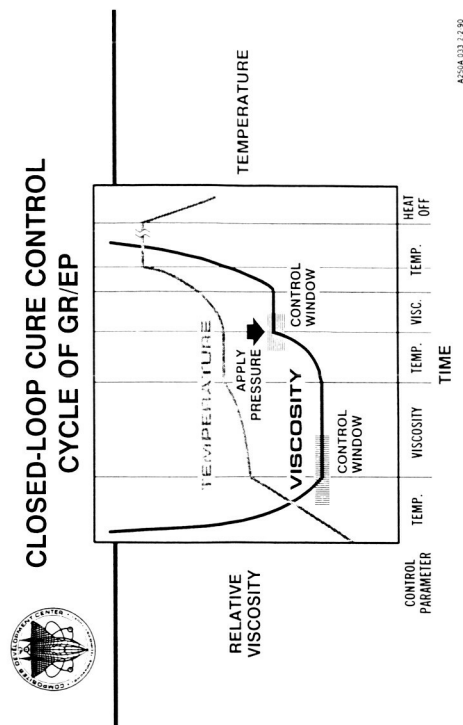


Fig.3 Closed-Loop Cure Control Cycle of Graphite/Epoxy Composites

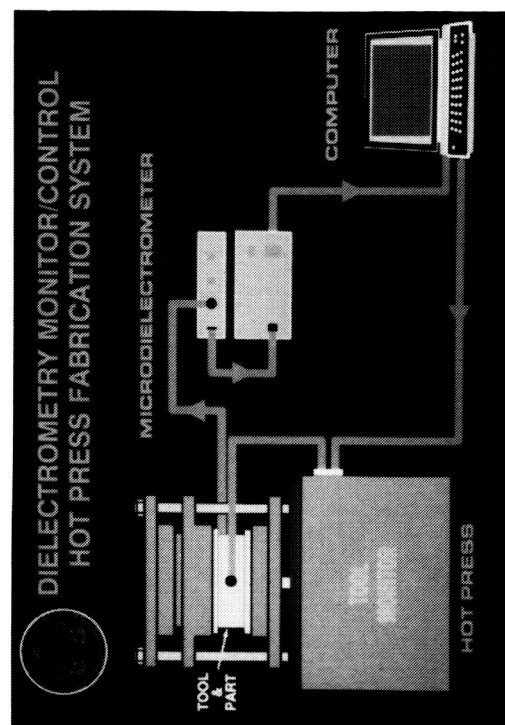


Fig. 4 Dielectrometry Monitor/Control Hot Press Fabrication System

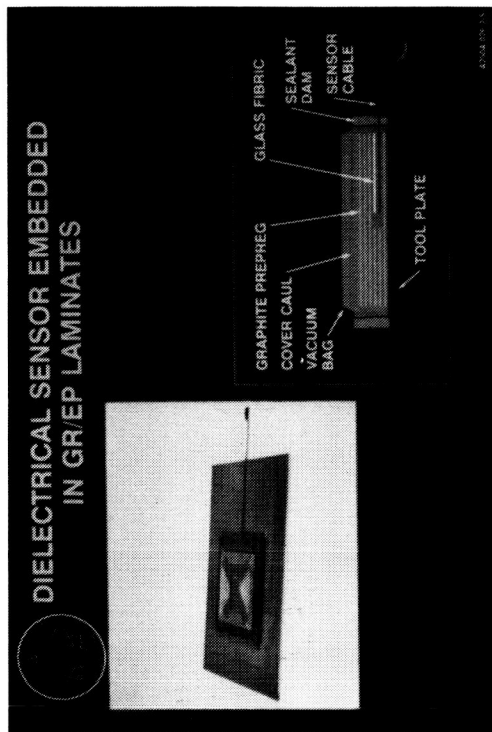


Fig. 5 Dielectrical Sensor Embedded in Graphite/Epoxy Laminates

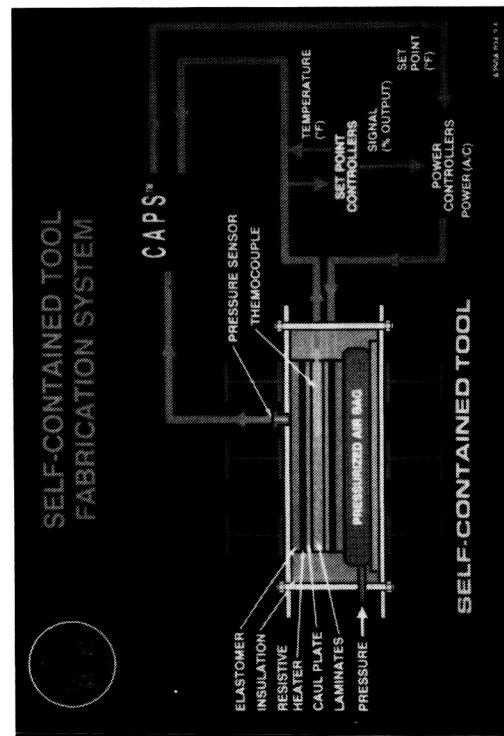


Fig. 6 Self-contained Tool Fabrication System

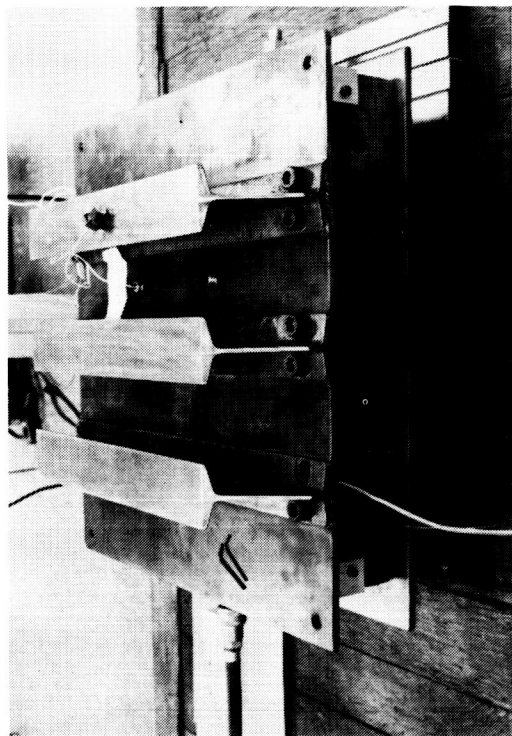


Fig. 7 Pressure Transducers and Self-Contained Tool

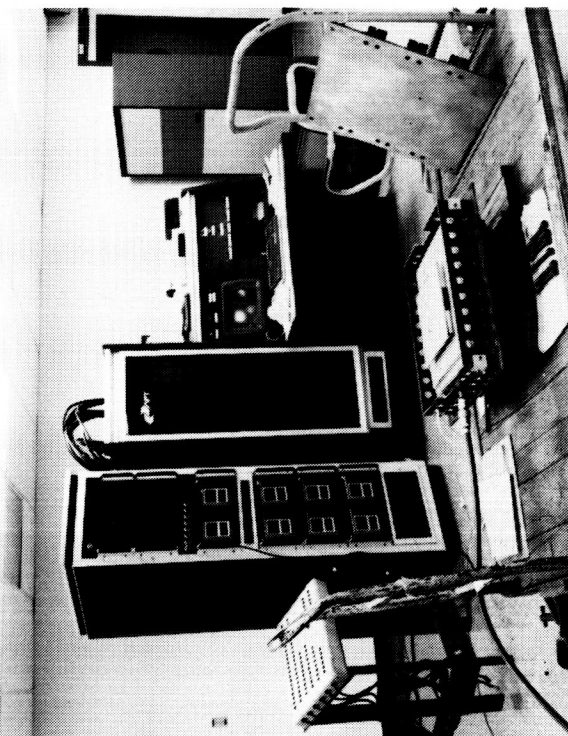


Fig. 8 CAPS, Power Unit, Sensors and Self-Contained Tool

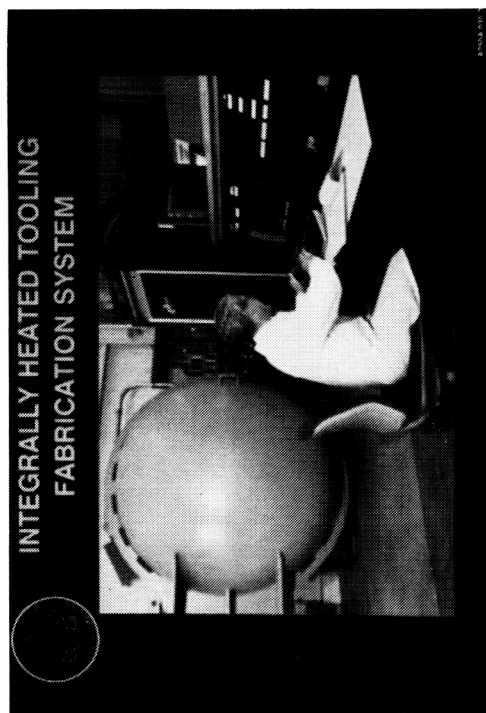


Fig.9 Computer Controlled Pressure Vessel Fabrication

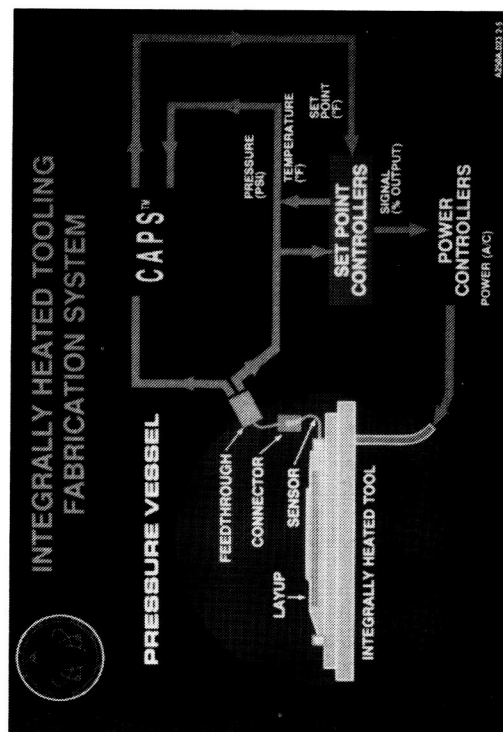


Fig. 10 Integrally Heated Tool and Pressure Vessel Fabrication System

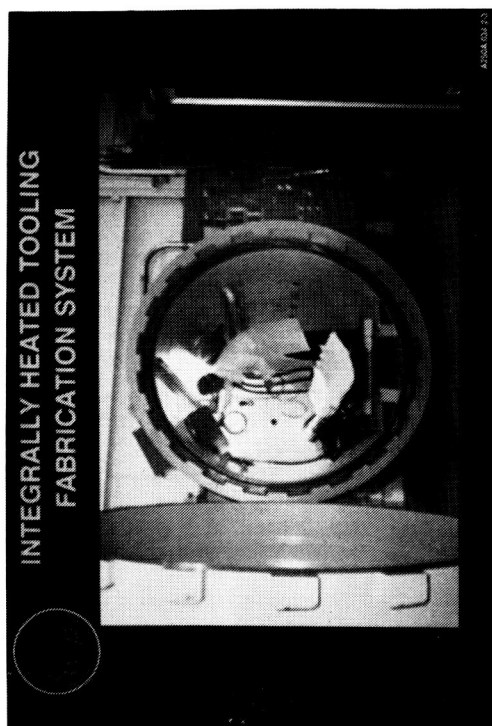


Fig. 11 Integrally Heated Tool and Pressure Vessel Fabrication System

## DEVELOPMENT OF A LOW-COST, MODIFIED RESIN TRANSFER MOLDING PROCESS USING ELASTOMERIC TOOLING AND AUTOMATED PREFORM FABRICATION

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### SUMMARY

This paper describes the design and process development of low-cost structural parts made by a modified resin transfer molding process. Innovative application of elastomeric tooling to increase laminate fiber volume and automated forming of fiber preforms are discussed, as applied to fabrication of a representative section of a cruise missile fuselage.

### INTRODUCTION

Advanced composite materials have found aerospace applications in which they help meet performance requirements of strength, weight, and radar signature reduction, although historically at very high production costs. Cost reduction is required if composite materials are to remain viable candidates for high-performance applications.

A low-cost structure development project was initiated as part of a study of ways to reduce the cost of cruise missiles. An integrated product development team was formed with representatives from design, manufacturing, materials, stress analysis, and cost estimating disciplines. A conceptual generic vehicle was created and is shown in F1. The team decided to evaluate composite structure for the midbody section of the vehicle and to develop a study configuration to aid in composite material process development. Guidelines for low cost were developed, such as fabricating large integral assemblies, integrating automation, providing low process cycle time, and eliminating bolted joints.

Trade studies were conducted on composite designs using filament winding, pultrusion, thermoforming, and resin transfer molding (RTM) processes. Cost, functional, and producibility evaluations led to the selection of the RTM process as the lowest-cost process with the greatest versatility. This versatility is illustrated by the many different features provided by the process, including the ability to make large integral assemblies and net shape moldings. Also, a wide variety of structural parts can be formed using the same design, process, and tooling technology. Our team chose to investigate the RTM process and to develop process modifications to increase its usefulness. The RTM process is generally applicable to all aerospace structural fabrication. For the purposes of this study, we chose to use a generic cruise missile structure.

This paper discusses the use of elastomeric tooling to provide a method of increasing fiber volume, resulting in a higher-quality laminate; a method of fabricating fiber preforms using automation; and design and fabrication of a demonstration section of an integral composite structure for a cruise missile midbody.

### RUBBER-ASSISTED RTM

Elastomeric tooling was investigated as a method to increase fiber volume and laminate quality over conventional RTM methods. Rubber-assisted resin transfer molding (RARTM) relies on



placing a rubber insert into the tool that expands a predetermined amount during cure, providing pressure to compact the fiber reinforcement. Comparison of the RTM and RARTM processes shows improvements in fiber volume and producibility. Our team has used the RARTM process to fabricate flat panels for material properties tests, stiffened panels for three-point bending tests, and skin stiffeners for the fuselage demonstration section.

Conventional RTM is a closed-mold, low-pressure process (R1). F2 is a diagram of the process. A dry fiber reinforcement called a preform is placed in a sealed tool cavity. The tool with preform and the resin to be injected is heated to a moderate temperature such as 130°F. After vacuum is applied to remove air from the tool cavity, the resin is transferred at low pressure. The temperature is then raised to the cure temperature with the resin remaining under pressure. The part is cured and then removed from the mold.

The RARTM process is shown in F3. The process is the same as conventional RTM except for the novel use of trapped rubber in the mold and a pressure relief valve to vent excess resin at the appropriate time in the process cycle. The rubber is sized (R2) to allow a low fiber compaction during the resin transfer stage of the process. This allows improved resin flow with better fiber wetting at a lower transfer pressure. After the resin is transferred into the preform in the tool cavity, the temperature is raised to the cure temperature. During this temperature rise, the rubber expands, due to its high coefficient of thermal expansion, and compacts the fiber reinforcement while squeezing out excess resin. A relief valve allows the excess resin to vent from the tool while maintaining adequate backpressure. Resin backpressure is required to produce a high-quality, low-void-content part.

An evaluation of the RTM and RARTM processes can be made by contrasting the processing parameters needed to produce comparable laminates. The processing parameters of interest are resin transfer pressure, resin transfer time, and total cycle time. "Comparable laminates" refers to fiber volumes of 65% for the finished laminates. The temperature and pressure cycles for the RTM process are shown in F4. For RTM, the fiber volume of the preform in the tool cavity at resin transfer is 65% because the tool cavity remains unchanged throughout the process. This results in a resin transfer pressure of 50 psi, a resin transfer time of 165 minutes, and a cycle time of 340 minutes. Temperature and pressure cycles for the RARTM process are shown in F5. The fiber volume of the preform in the tool cavity is 50% at resin transfer. This is less than for the RTM process and is possible because the rubber insert can be sized for easy resin transfer. After resin transfer and during the cure cycle, the rubber expands, forces out excess resin, and compacts the laminate. The resin transfer pressure of 20 psi, a resin transfer time of 25 minutes, and a total cycle time of 205 minutes are improvements over the conventional RTM process. T1 lists material properties for an RARTM panel at 66.9% fiber volume. This fiber volume is achievable with the conventional RTM process but with greater processing pressures and cycle time. The reduction in cycle time provides cost savings in the form of reduced labor and better tool utilization.

We have used the RARTM process to fabricate mechanical property test panels, tee-stiffened test coupons, and the skin stiffener in the fuselage demonstration section. These tools are shown with their rubber tool inserts in F6. In the case of tee and skin stiffeners, the rubber inserts have been shaped with a draft angle and made in two halves to facilitate loading the fiber preform into the tool cavity.

## FIBER PREFORM FABRICATION

A low-cost method of forming fiber preforms was developed. Using a powdered epoxy resin product, called a tackifier, a consolidated flat sheet of reinforcing plies is fabricated at low temperature and pressure. The resin secures the plies together so they can be cut to shape on an automated cutter. Forming takes place over a tool form at the same low temperatures and pressures used in the consolidation step. Patterns and parts have been fabricated using this method with fiberglass and graphite reinforcements and RTM. F7 is a schematic representation of this sequence.

The tackifier is a fully reactive, granulated epoxy resin powder with no hardener and a melting point of 140F. The tackifier is applied to individual dry fiber plies in a quantity that does not exceed 5% of the resin weight of the final part. The desired thickness of plies is built up in this manner and consolidated in an oven at 140°F under vacuum. This process lightly secures the plies to each other and facilitates the cutting and forming operations. The operation of fabricating these consolidated preform modules can be automated.

A trial pattern (F8) was designed to test the feasibility of cutting the sheet preforms using an automated ply-cutting machine. The pattern consists of 10 plies of 181 style fiberglass at 0-deg orientation. The designer generated the pattern on a computer-aided design (CAD) workstation and electronically transferred it to the cutting system computer. This procedure eliminates the dimensional errors of making a template from a drawing and using the template for cutting the pattern. The test pattern was cut without problems. The plies maintained their adhesion to each other at the edges, fiber fraying at cut edges was minimal, and all corner radii down to the 0.25-inch inside corner radius were cut smoothly. Sharp corners, such as those at the corners of the squares, did result in a small cut into the sheet where the blade traveled beyond the corner. This type of corner would not be used, however, due to possible stress concentrations. There was no accumulation of tackifier material on the cutting blade.

The tackifier material also allows forming patterns cut from the sheet preforms. Vacuum pressure and heat, at 130° to 140°F, are applied to form the pattern over a tool form. The heat softens the tackifier to allow this forming but does not cure the resin material. The sheet preform can be reformed if necessary. Preforms made in this manner can be stored for long periods at room temperature. A pattern, tool form, formed pattern, and completed RTM frame are shown in F9.

A benefit of the preform fabrication process is consolidation of the plies, which provides easier tool closing and results in a part with a lower void content. The tackifier dissolves and reacts with the resin transferred through the reinforcement during the RTM process. Both fiberglass and graphite preforms have been fabricated with this method and used to fabricate parts with both conventional RTM and RARTM processes.

## DEMONSTRATION ARTICLE

A stiffened/skin subassembly was designed to demonstrate the benefits of large parts, integral assemblies, and the advantages of the modified RTM process described above. The subassembly is a section of the midbody design used in our trade studies. Composite tools were designed and fabricated with a unique progressive resin transfer technique. Fabrication of the stiffened/skin subassembly was completed successfully.

A composite material midbody was designed for the generic cruise missile baseline vehicle using the RTM process; see F10. The design approach consists of upper and lower stiffener/skin subassemblies secondarily bonded to a separately fabricated RTM center structure. This two-part

structural approach – skin subassemblies and center structure – was chosen to avoid complex enclosed cavities that would require segmented tooling or dissolvable mandrel tooling. The approach also provides for easier inspection of the parts.

The stiffened skin subassembly is shown in F11, along with the design of the mating structure: the wing cavity floor panel. The development section is 30 inches long and has a radius of 12 inches. The skin consists of four modules of graphite cloth. The stiffeners are S-2 fiberglass cloth. The secondary bond between the skin and the floor panel is a stepped lap joint. A 350°F curing epoxy resin is used.

The subassembly tooling is laminated graphite and fiberglass epoxy, as shown in F12. An outer shell stiffened with support frames forms the smooth outside surface of the skin. The inner shell is made with cavities for rubber inserts at the skin stiffener locations. The two shells are bolted together and sealed with an O-ring seal. Combination resin injection and vent ports are provided at three levels in the tool.

Fabrication of the subassembly begins with fabrication of glass preforms for the skin stiffeners. The skin stiffeners are formed by placing several fiberglass angles back to back. The fiberglass plies are formed into preform sheets three plies thick, cut, and then formed into an angle over a tool form at 140°F with vacuum pressure. The angles are assembled into the rubber inserts and placed into the tool. Next, preparation of tackifier graphite cloth preforms for the skin is completed. These are made in modules of four plies. The modules are placed on the inner shell over the tee preforms and inserts. Four modules are stacked to provide the 0.25-inch skin thickness required. A problem developed with wrinkles as the modules were formed over the 12.0-inch radius of the inner tool. Due to time constraints, a solution was not worked out at the time and the plies were laid down on the tool individually. Techniques to eliminate the wrinkling problem are in progress and involve using a line or grid application of the tackifier rather than a broad area application and fewer plies in a module. A layer of cured radar-absorbing material (RAM) material is placed on the outside surface of the skin. The tool is then closed, bolted together, and leak-checked.

The resin transfer takes place at a temperature of 120°F and a pressure of 20 psi. Vacuum is drawn on the part before resin transfer. With the tool standing on one end, the resin is transferred by a batch pressure pot pumping system at the low end of the tool through three ports. To avoid problems with loss of resin pressure through the length of the preform, dual resin inlet and vacuum ports are placed at the middle of the tool and near the top end. As resin appears at these dual ports, the vacuum line is closed off and resin is transferred into the tool at that point. This method of progressive resin transfer aids in fabrication of large parts. Total injection time for the 23.9-lb subassembly was 180 minutes, including a 40-minute purge time.

Lessons learned from design and fabrication of the demonstration article (F13) include:

- A two-part structural approach simplifies tooling and inspection.
- Progressive resin transfer aids resin transfer molding of large integral assemblies.
- Composite tools contribute to shorter part cycle times through faster heating and cooling rates.
- Vacuum and positive pressure leak checks should be performed.
- RARTM can reduce tool closing forces and resin injection pressures.

## CONCLUDING REMARKS

The RARTM process allows a low-cost process to compete with more expensive processes in fiber volume and laminate quality comparisons and provides improvements in processability over the conventional RTM process. These improvements are lower resin transfer pressures, shorter process cycle time, easier tool closing forces, and simplified part removal from the tool.

The method of preform fabrication described offers benefits that reduce the cost of the RTM process. Individual cutting and placing of plies is eliminated by a method that allows full or partial automation at the sheet stage, pattern cutting, and forming operations. Hand labor and the problems of securing dry plies in a shaped mold are also eliminated. Additionally, the use of the tackifier resin provides preforms that are preconsolidated, aiding closing of the tool.

Producibility advantages have been shown through fabrication of the demonstration section. Large integral assemblies incorporating different structural features, graphite and fiberglass reinforcements, and a radar-absorbing coating can be fabricated using RARTM and consolidated preforms. RARTM, the progressive resin transfer method, and composite tooling provide lower part cycle times through lower resin transfer pressures, maintaining transfer pressure, and faster tool heating and cooling.

Composite fabrication costs can be reduced through the use of low-cost processes that produce parts with a high fiber volume, use automation in making preforms, and allow fabrication of large net shape moldings integrating many parts. The resin transfer molding process was chosen as the lowest cost process to produce composite structure for cruise missiles. Our trade studies have developed advanced composite structures that meet the requirements of the next-generation cruise missile and can be produced at a cost of \$60 per pound. The conventional RTM process can be improved upon by using elastomeric tooling in the RARTM method and by the automated fabrication of fiber preforms.

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Table 1. Material properties for RAKTM.

| Property                                                         | Average Value        |
|------------------------------------------------------------------|----------------------|
| Fiber volume                                                     | 66.9%                |
| Resin content weight                                             | 26.8%                |
| Tensile strength                                                 | 553.9 MPa (80.3 ksi) |
| Tensile modulus                                                  | 55.8 GPa (8.09 msi)  |
| Tensile strain at failure                                        | 9.767 $\mu\epsilon$  |
| Compressive strength                                             | 371.7 MPa (53.9 ksi) |
| Compressive modulus                                              | 53.0 GPa (7.68 msi)  |
| Compressive strain at failure                                    | 7.467 $\mu\epsilon$  |
| Interlaminar shear strength                                      | 52.0 MPa (7.55 ksi)  |
| Flexural strength                                                | 621.4 MPa (90.1 ksi) |
| Flexural modulus                                                 | 37.0 GPa (5.37 msi)  |
| Graphite - AS4 6K SHS, 0.15ply [0, +45, -45, 0, 90] <sub>s</sub> |                      |
| All properties in longitudinal direction                         |                      |

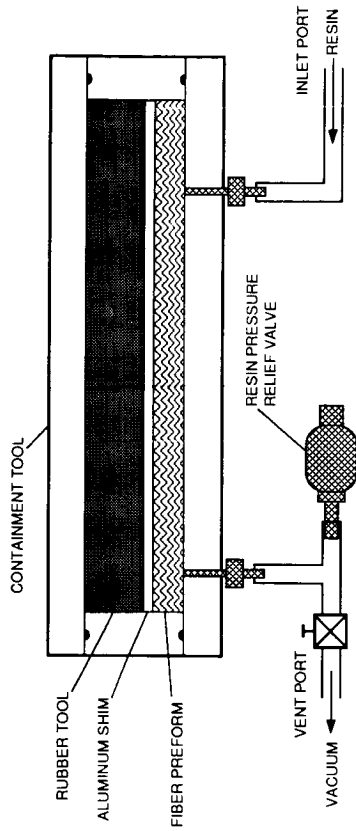


Figure 3. Rubber-assisted RTM diagram.

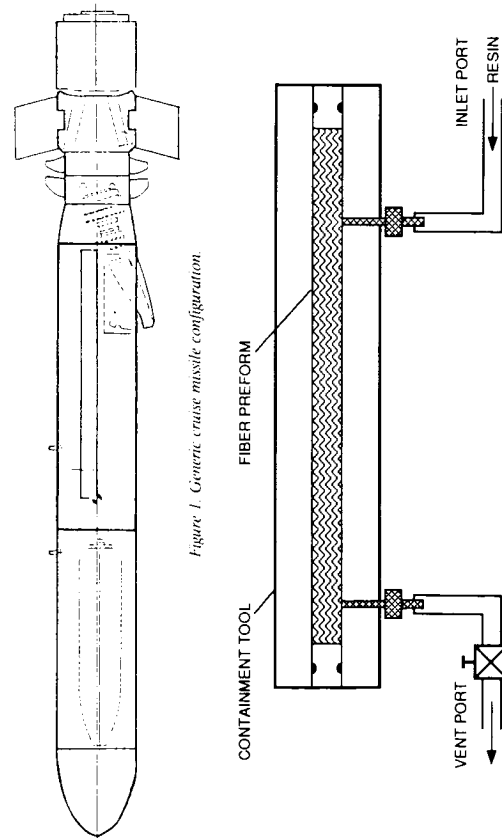


Figure 2. Conventional RTM process diagram.

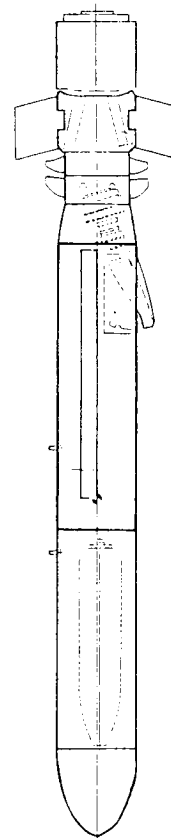


Figure 1. Generic cruise missile configuration.

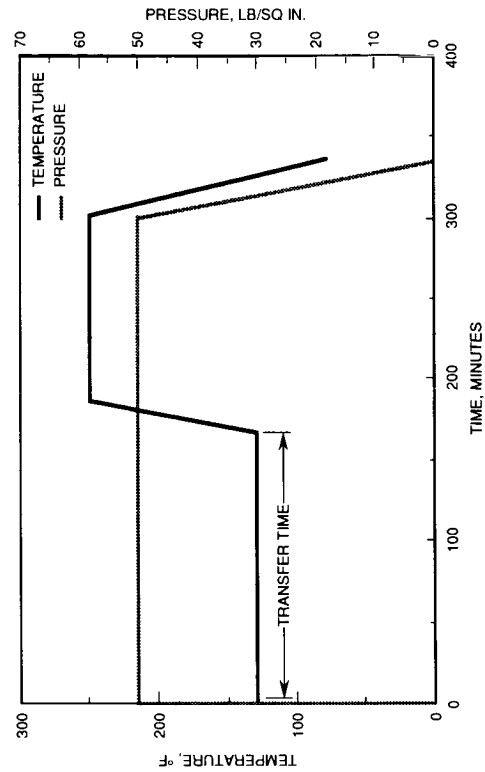


Figure 4. RTM pressure, temperature, and time cycles.

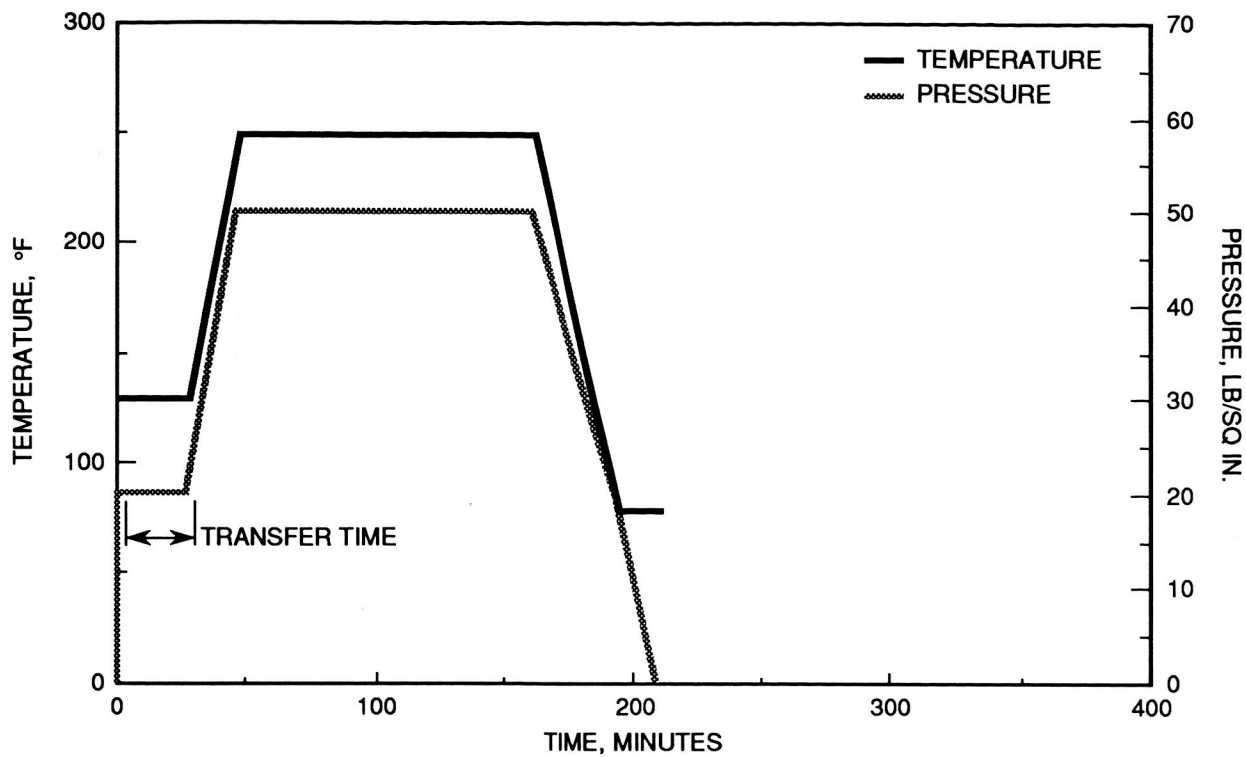


Figure 5. RARTM pressure, temperature, and time cycles.

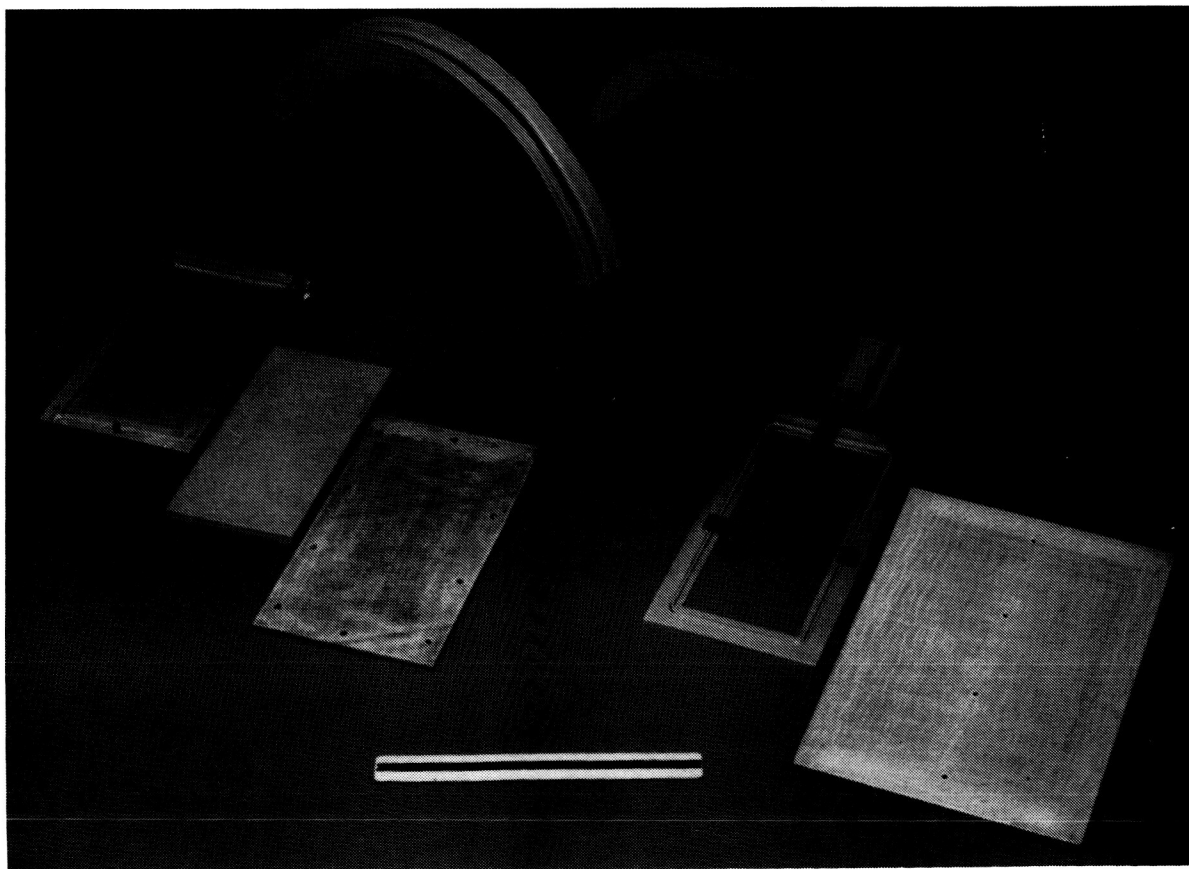


Figure 6. RARTM tools, mechanical property panels, tee stiffeners, and skin stiffeners.

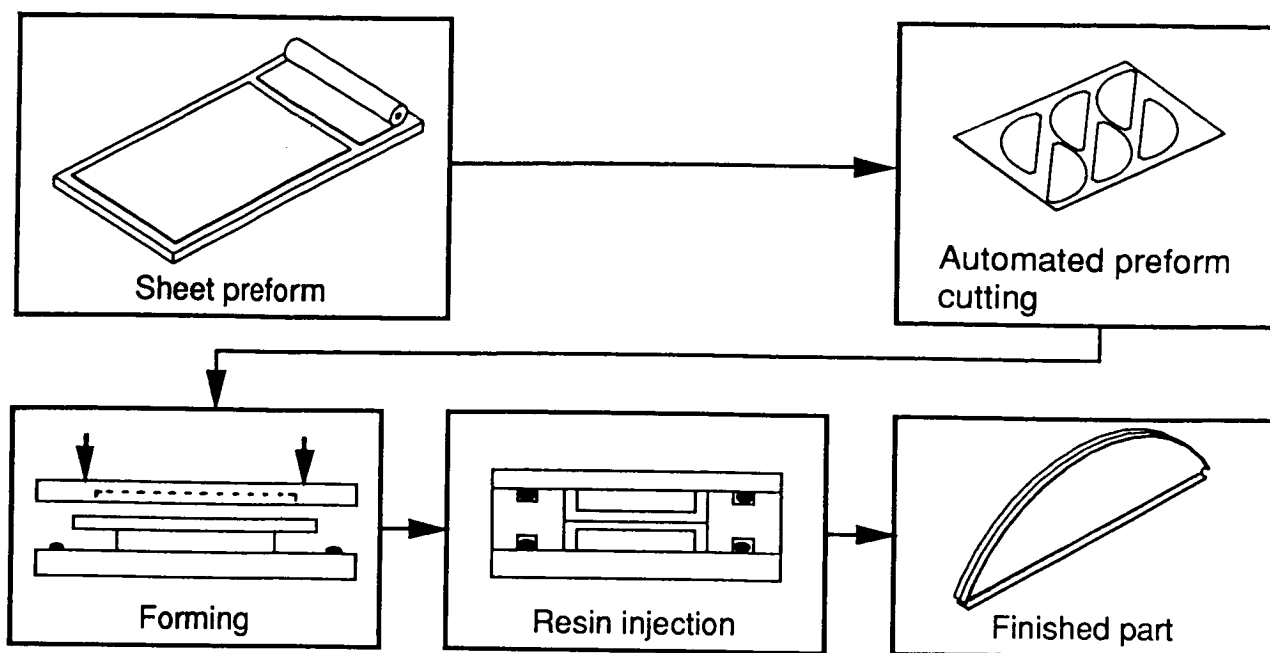


Figure 7. Automated preform fabrication sequence.

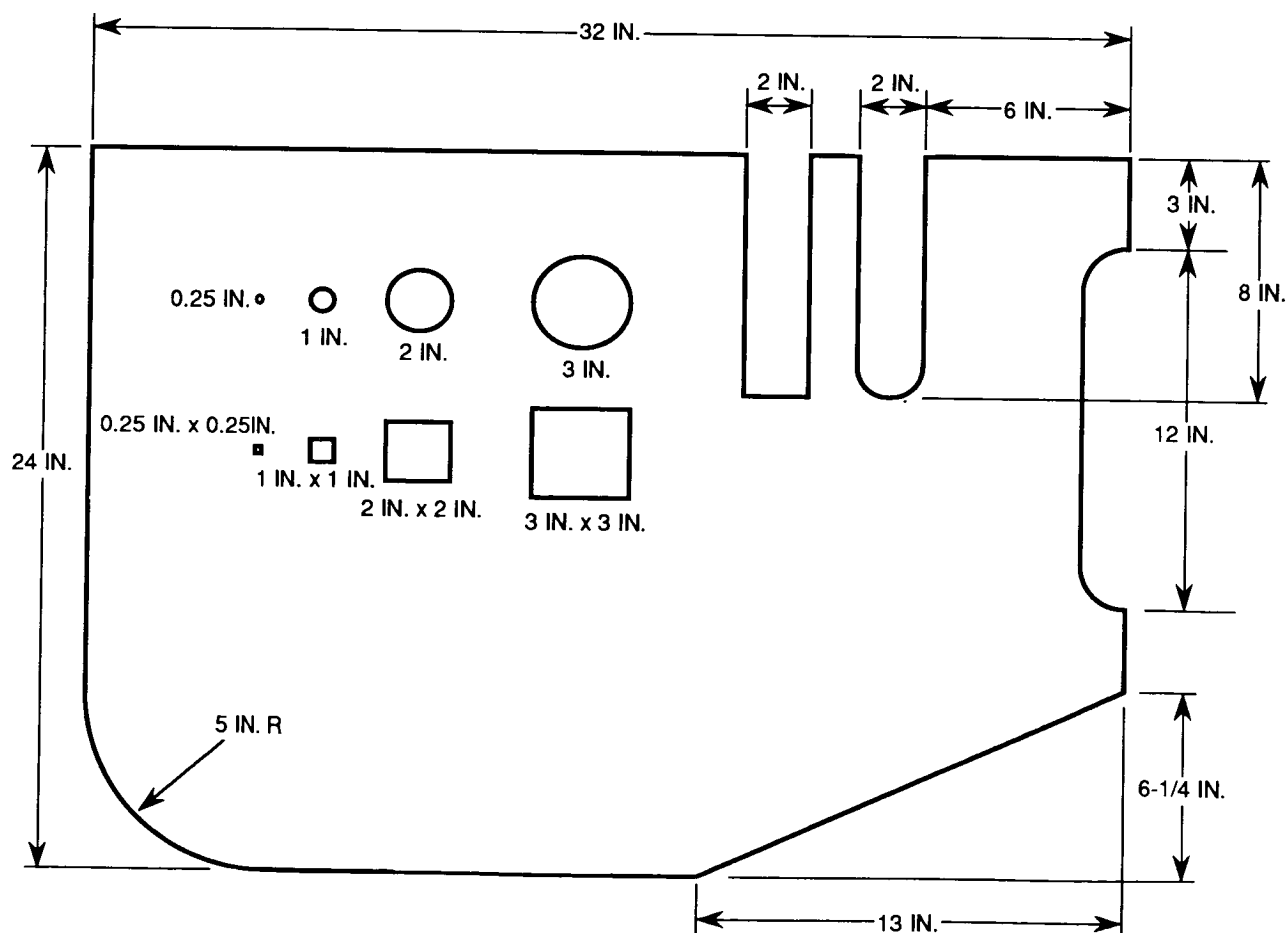
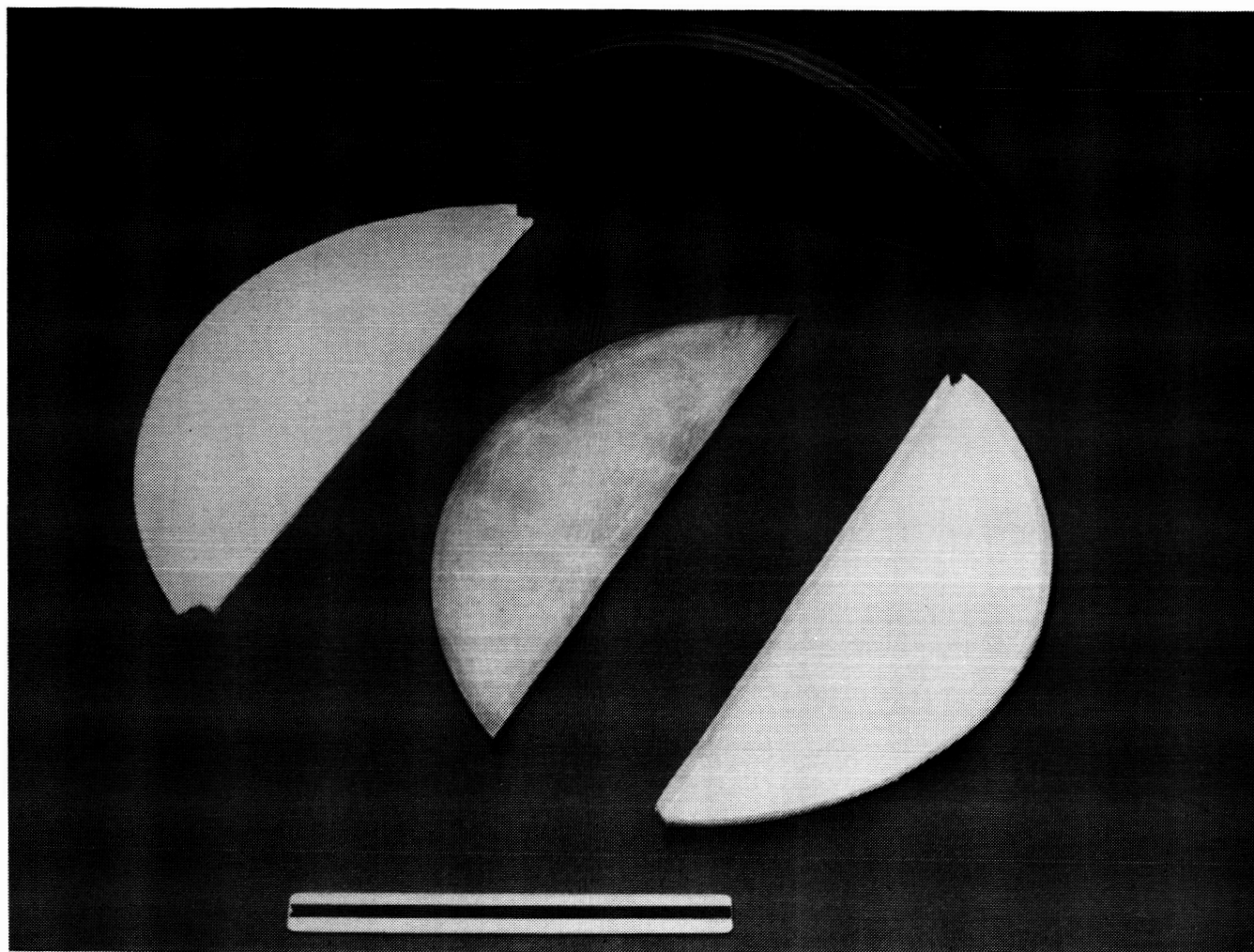
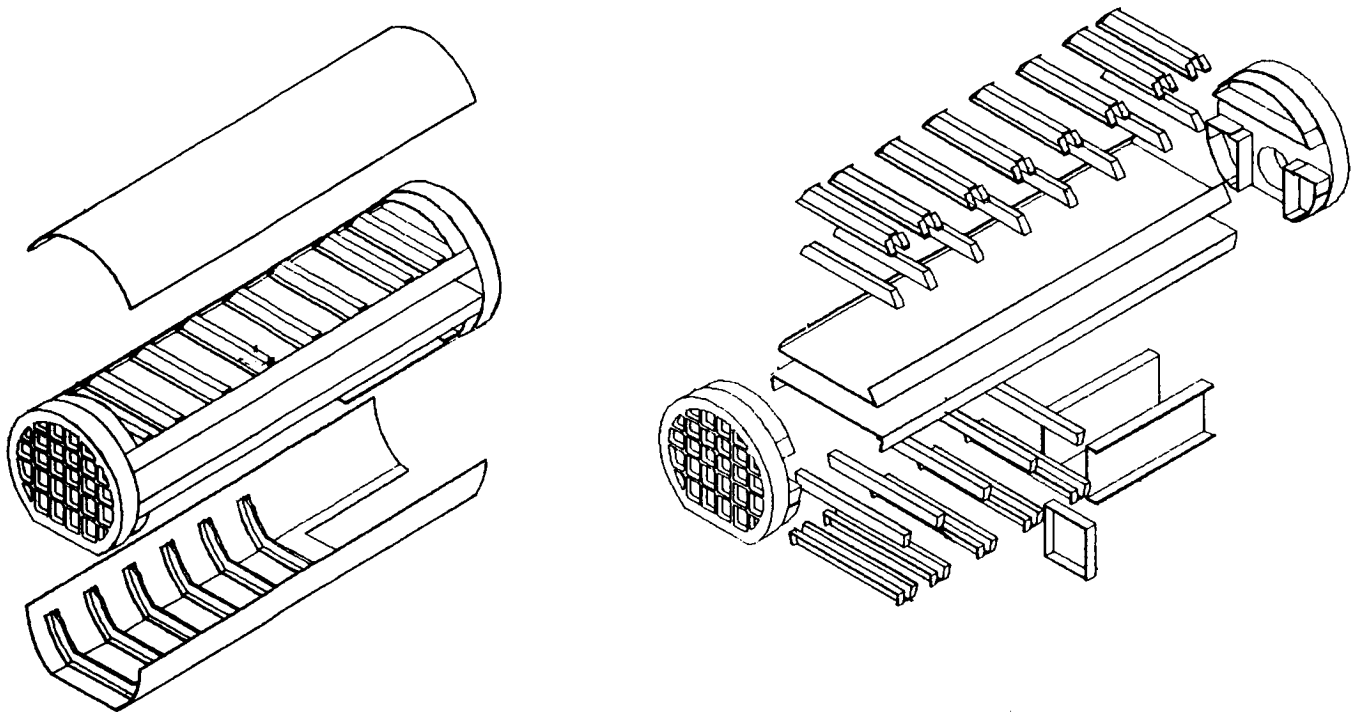


Figure 8. Test pattern for automated preform cutting.

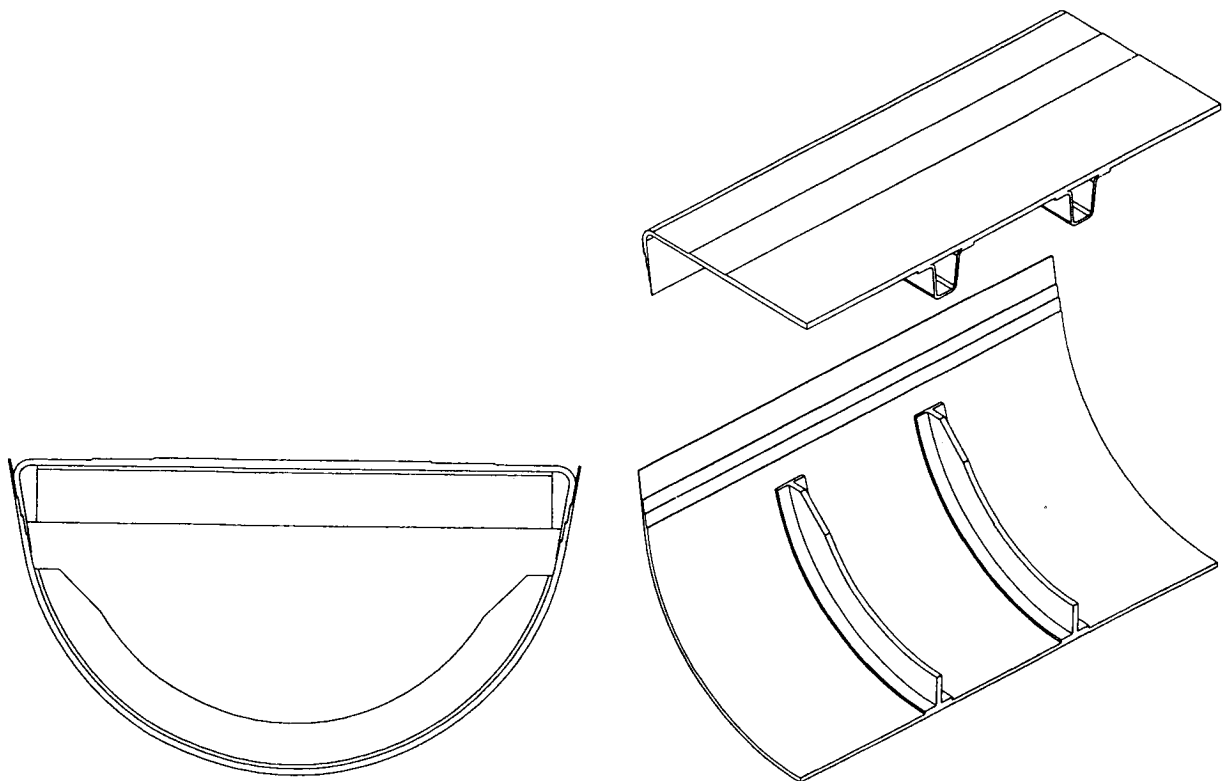


*Figure 9. Forming operation for fiber preforms.*





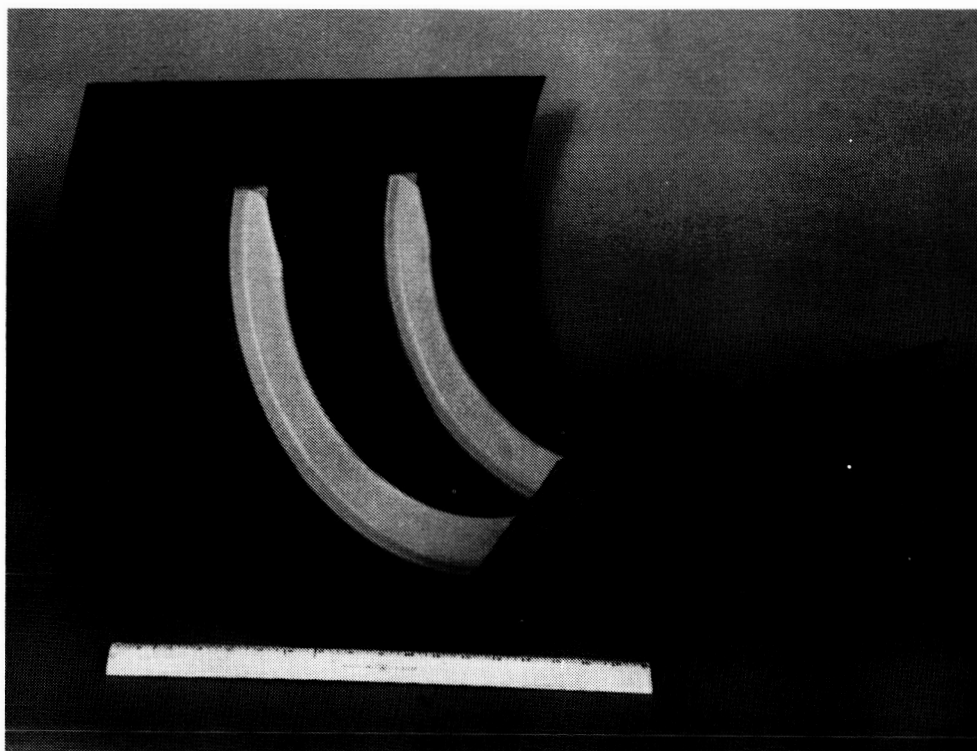
*Figure 10. RTM midbody design configuration.*



*Figure 11. RTM midbody demonstration article design.*



*Figure 12. RTM midbody demonstration article tooling.*



*Figure 13. RTM midbody demonstration article.*

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## "Design of Fabric Preforms for Double Diaphragm Forming "

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### Summary

Resin Transfer Molding (RTM) has the potential of becoming one of the most cost effective ways of producing composite structures since the raw materials used, resin and dry fabric, are less costly than prepregs. Unfortunately these low material costs are offset by the high labor costs incurred downstream to layup the dry fabric into 3 dimensional shapes. To reduce the layup costs, double diaphragm forming is being investigated as a potential technique for creating a complex 3D preform from a simple flat layup.

As part of our effort to develop double diaphragm forming into a production capable process, we have undertaken a series of experiments to investigate the interactions between process parameters, mold geometry, fabric weave, tow size, and the quality of the formed part. The results of these tests will be used to determine the forming geometry limitations of double diaphragm forming and to characterize the formability of fabric configurations. An important part of this work has been the development of methods to measure and analyze fiber orientations, deformation angles, tow spreading, and shape conformation of the formed parts. This paper will describe the methods used to mark the plies, the double diaphragm forming process, the techniques used to measure the formed parts and the calculation of the parameters of interest. The results can be displayed as 3D contour plots. These experimental results have also been used to verify and improve a computer model which simulates the draping of fabrics over 3D mold shapes.

### Introduction

One of the major obstacles limiting the expanded use of composite materials is their relatively high cost in comparison to alternative materials such as metals. These costs are attributable to expensive raw materials as well as labor intensive manufacturing

processes. To develop new processes which reduce the cost of manufacturing composite structures, it is important to understand first the costs in current manufacturing processes. However, this cost-based design approach presents a problem, because the costs of composites manufacturing processes are not well understood. In addition, there are numerous combinations of raw materials and processing techniques that can be combined to manufacture a particular part each with its own structural and cost benefits. Due to the variety and complexity of these processes, it is important to use cost analysis tools in the early stages of process development efforts [1]. By using cost analysis tools, not only can the most cost effective current processes be identified but the cost centers for the processes can also be located. The results of such analysis can then be used to target and to set specifications for the development of new improved processes.

Cost analysis has shown that Resin Transfer Molding (RTM) holds promise for cost effective composite structure manufacturing since the raw materials required, dry fabric and resins, are less costly than prepregs [2]. These low material costs are currently offset, however, by the labor intensive downstream operations required to layup the three-dimensional fiber preforms. To reduce RTM preform layup costs, double diaphragm forming is being investigated as a cost effective technique for producing three-dimensional preforms.

In this paper, we will first describe the cost motivation for development of the double diaphragm preforming process. Then we will describe a set of methods that are being used to measure and analyze fiber orientations, deformation angles, tow spreading and shape conformation of formed parts. Finally we will show how the results from experiments based on the described methods were also used to verify and improve a computer simulation that drapes fabrics over 3D contours. This simulation will serve as a basis for a design tool that will help predict fiber orientations and other important properties in formed parts.

### **Cost Analysis**

In this section we will describe the results of the cost analysis that motivated the development of the double diaphragm preforming process. In the initial cost study, several state-of-the-art manufacturing processes, including filament winding, automatic tape laying, thermoplastic press forming and resin transfer molding, were compared. (See [2] for details of the study.) For the specific family of part geometries considered

in the study, resin transfer molding was the most cost effective process examined at all production volumes and was chosen as a starting point for further process development.

RTM's main advantages over the other processes considered in the study are that it uses the lowest cost constitutive materials possible (fiber and resin), it has a relatively short cycle time, and that a high degree of geometric complexity and part integration is possible. Also, because injection pressures are low (10-80 psi), equipment costs are typically small (~\$25-50K).

A breakdown of RTM cost components is shown in Figure 1. This data was obtained by averaging an empirical database of actual costs for a number of RTM parts currently in production in low and mid-volume aerospace applications.

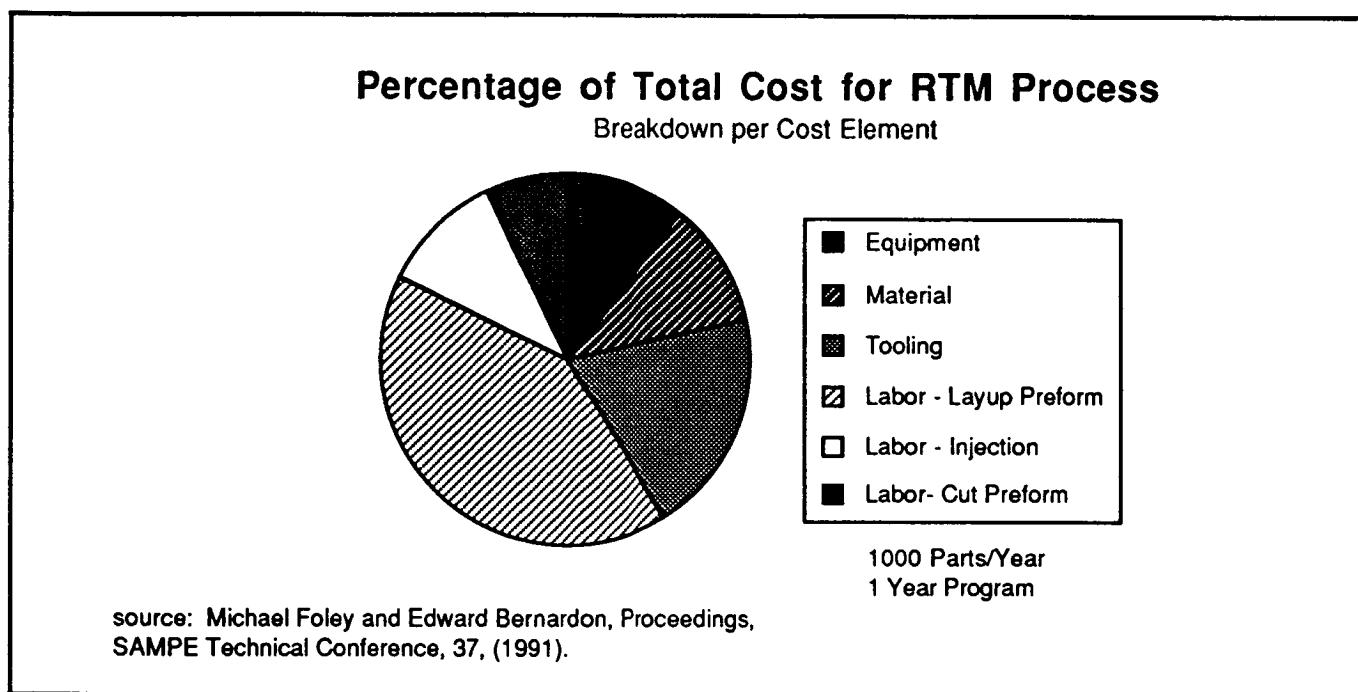


Figure 1

Figure 1 indicates that the largest percentage ( $\approx 40\%$ ) of the cost of the traditional RTM process lies in preparation of the fabric preform. The trends identified here are based on parts of small to medium size and of low to medium complexity. The contribution of major cost components such as preform preparation will become even larger as parts become larger and more complex.

As a result of this analysis, it became clear that to improve the cost effectiveness of RTM, alternative methods of preform preparation, such as double diaphragm forming, must be investigated.

## Double Diaphragm Forming

A schematic of a double diaphragm former for preform preparation is shown in figure 2. The two-dimensional dry fabric preform is trapped between two elastomeric diaphragms mounted in frames. A vacuum is drawn between the diaphragms through a breather strip to hold the preform during the forming process. The preform/diaphragm sandwich is situated over a vacuum forming chamber containing the part mold. A vacuum is drawn inside the chamber to shape the preform diaphragm sandwich to the mold.

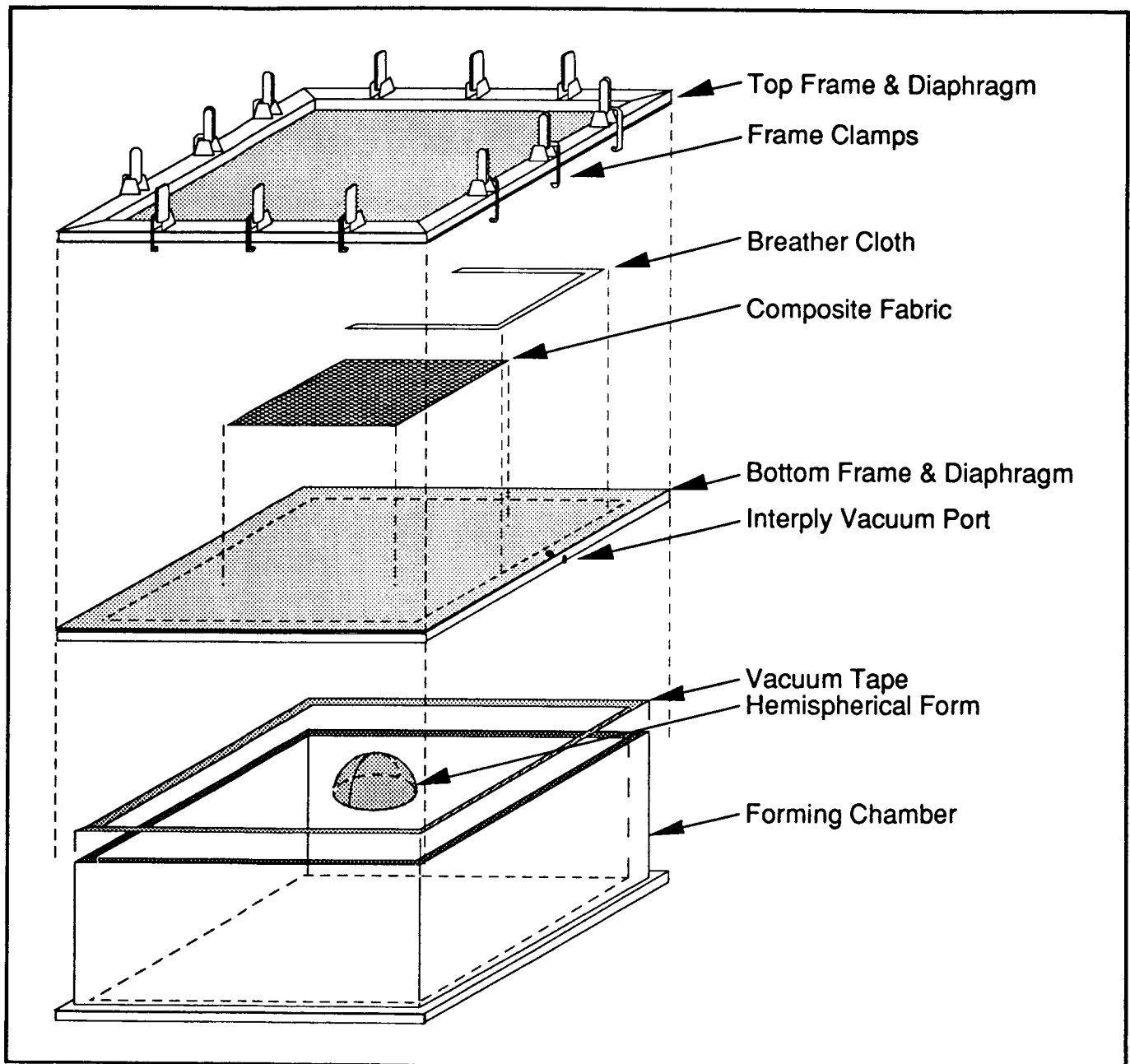


Figure 2. Double Diaphragm Former

The analysis described above indicated the primary RTM cost component is the three-dimensional layup of the fiber preform. The greatest advantage of the double diaphragm forming process is that all preform layup is completed in the flat, thus eliminating costly three-dimensional layup. Since the time required for flat layup is small relative to three-dimensional layup, layup labor costs are greatly reduced. In addition, automated techniques for preparing flat layups are more easily developed than those for complex 3D geometries and the diaphragms can be used to support plies during transport. The diaphragms maintain constant contact with the preform, inhibiting out of plane buckling or wrinkling, and because the tool is not in contact with the fabric, the preform can be removed easily after forming. As a result, this three-dimensional forming process is very repeatable and tuning preform shapes can significantly reduce post trimming operations, thereby increasing material utilization and reducing scrap costs.

Double diaphragm forming has several advantages over single diaphragm forming and hand layup since the two diaphragms inhibit out of plane buckling and wrinkling thus improving part formability. The diaphragms can also support plies during transport and the tool is not in contact with the fabric facilitating the removal of the part after forming.

Although double diaphragm forming reduces preform layup time, the complexity of preforms that can be manufactured are not as geometrically complex as those possible with hand layup. The geometric limits of the forming process are a function of the former design (diaphragm stiffness, elongation, etc.) as well as properties of the preform itself. To understand and extend the complexity envelope of the double diaphragm forming process, experimentation has begun to investigate the effects of various preform material properties on the forming limits of double diaphragm forming. As we better understand the forming capabilities of preform materials, the size and complexity of manufacturable parts will increase. As a result, smaller preforms could be consolidated, reducing not only handling during the layup process, but also eliminating preform joining operations.

## **Test Plan**

After the initial try out of the diaphragm forming equipment and determination of suitable process parameters, a series of experiments was performed using two different molds, a 4 inch hemisphere and a 4 inch cube with 1/2 inch corner radii, both illustrated in Figure 3. The four test materials were each draped in several orientations under



various initial conditions. Differences in formability as influenced by fabric weave, tow size and drape configuration were examined and quantified where possible.

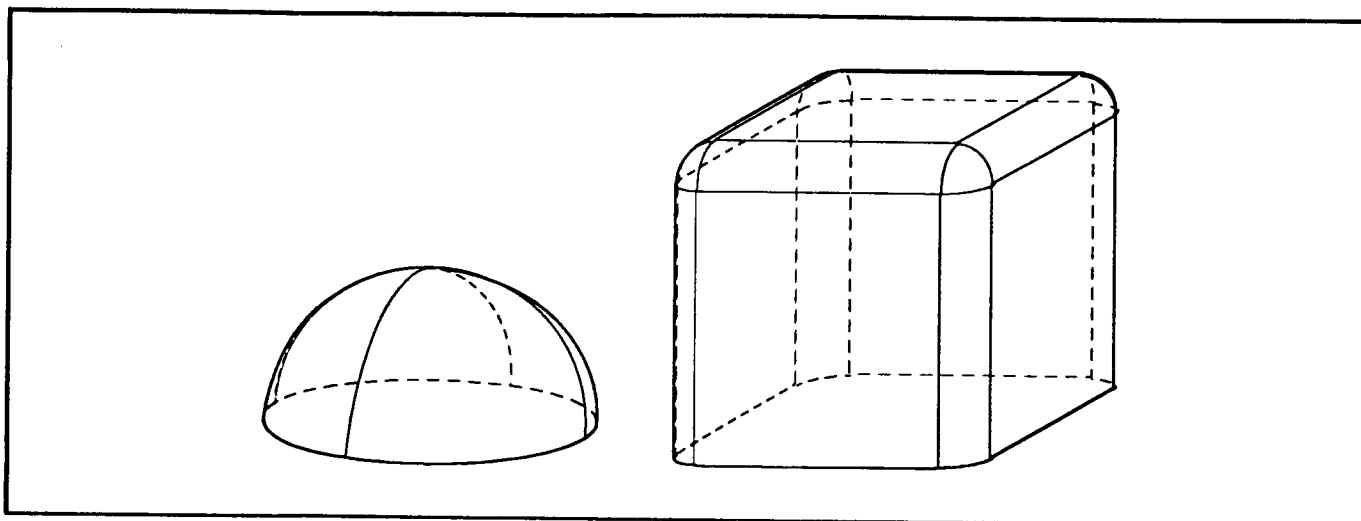


Figure 3. Composite preform mold shapes

To quantify the results, a method was devised to measure the deformation parameters of interest, including deformation angle, spreading and shape conformation. The results of these measurements can be displayed as 3 dimensional contour plots and compared directly to the results of the Draper simulation of the fabric lay up process.

The insights we make into the mechanics of diaphragm forming will guide the design of the next generation of formers. Although these tests are being performed on a double diaphragm former, they are contributing significantly to our knowledge of the deformation of woven fabric, including the interaction of part geometry and fabric material and weave.

### Experimental Procedure

The process starts by marking the composite fabric with a grid pattern that will later guide the measurement of the part. To achieve a uniform grid, a silkscreen with small dots in a 0.5" grid pattern was used to mark batches of material before testing. After several tests an ink and pattern were found which adhered to the materials but did not interfere with the forming process. After marking, the plies were cut to size, 8" square for the hemisphere and 12" square for the cube.

Next, the ply is placed on the bottom diaphragm and sprayed with a starch solution that will dry and hold the preforms shape. The starch solution contains poly-vinyl-alcohol

dissolved in hot water and diluted with isopropyl alcohol. The breather cloth is then positioned and the top diaphragm clamped in place.

To begin forming, a vacuum is pulled between the diaphragms, then a vacuum is drawn in the forming chamber which pulls the diaphragm-ply sandwich down over the mold. When forming has ceased, a bank of infrared lamps is used to drive some of the volatiles in the starch. After the ply has begun to dry, air is forced between the diaphragms, the top diaphragm removed, and the drying finished. After the preform is completely dry, the part can be removed and the vacuum in the forming chamber released.

The Laser Coordinate Measurement System (LCMS) developed at CSDL was used to digitize the grid points on each part, capturing the three dimensional coordinates. The grid dots were located by hand with a pressure sensitive laser probe. The coordinates were recorded in an ASCII character file and then converted into Patran neutral file format and viewed using Patran on a Sun Sparc station. The data was screened for correctness (i.e., removal of double data points) and the refined model was stored as a second Patran neutral file. This file was then passed through a program that calculated the deformation angle and x and y (or warp and weft) spreading.

The deformation angle is defined as the smallest included angle in the rectangle defined by four grid points. In Figure 4 these points are labeled A, B, C and D. The deformation angle for corner A is given by the dot product of two vectors AB and AD. The angle is calculated for each corner and the smallest angle is the deformation angle recorded for that quadrangle.

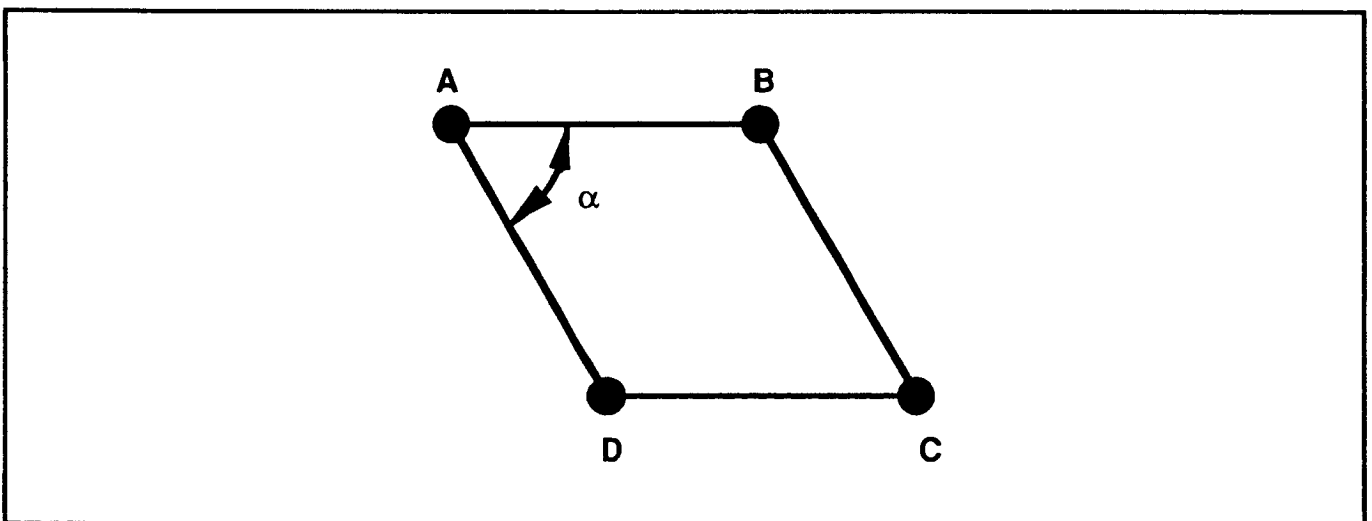


Figure 4. Definition of measured parameters

Spreading is defined as the ratio of the overall length of a side divided by the original length. The x and y directions correspond to the warp and weft of the ply which were orthogonal prior to forming. In figure 4 the X direction spreading would be the distance from point A to B divided by 0.5", the original distance between grid points.

After preprocessing the inspection data can be shown as a series of Patran generated 3D color contour surface plots of deformation angle and spreading in the x and y directions. An example plot of a formed hemisphere is shown in figure 5.

The digitized data can also be used to measure the degree of conformation to the desired shape. The distance from the digitized points to the ideal surface (that of the mold) can be determined and the results displayed in the form of a 3 dimensional contour plot. The volume of the space between the formed part and the mold could also be calculated. These techniques could result in effective quality control methods for double diaphragm and other types of preform forming.

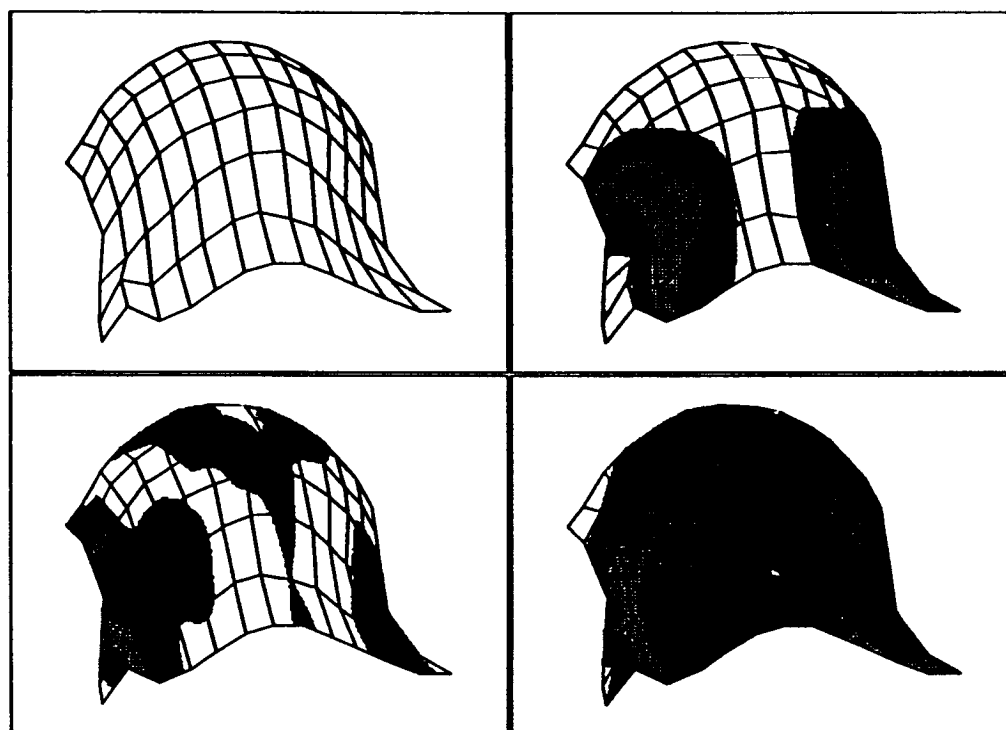


Figure 5. Four views of data derived by digitizing a grid of points on an experimental composite preform. From upper left, wire frame representation of data, contour plot of deformation angle, and contour plots of spreading in the x and y directions respectively.

## Simulation of fabric draping

A drape simulation code based on work done at the University of Delaware's Center for Composite Materials [3] was used to support the diaphragm forming experiments. Figure 6 shows the results from the drape simulation over a filleted hemisphere. When the simulation plot is scaled properly and compared directly with the experimental deformation angle plot in figure 5, the simulation is shown to have accurately predicted both the location and magnitude of fabric deformation. The simulation will be extended and modified based on the results of the experiments.

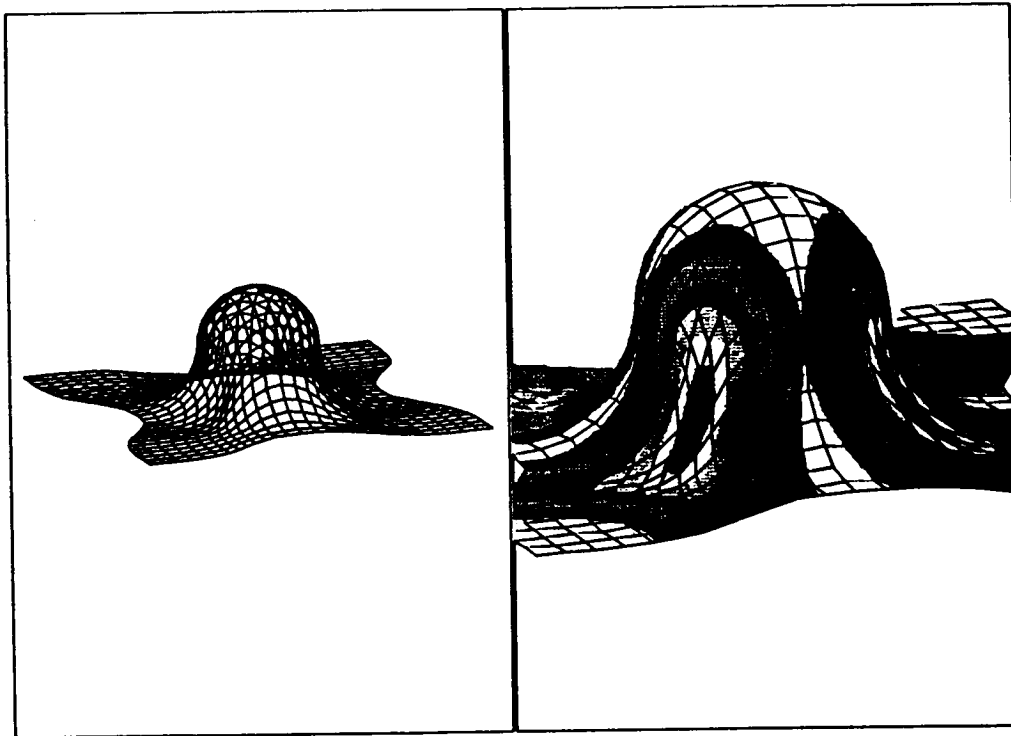


Figure 6. Two views of the results from the drape simulation over a filleted hemisphere. On the left, a wire frame representation of data, on the right a contour plot of deformation angle.

## Conclusion

Resin transfer molding (RTM) is a very promising and potentially cost effective manufacturing process for complex structural composite parts. However, most of possible cost savings are currently not realized because of the great expense involved in hand layup of individual composite plies on three dimensional forms. We have begun to investigate double diaphragm forming as a possible means to readily produce 3D composite preforms from 2D ply stacks. These preforms could then be quickly and

easily incorporated in an RTM mold, thus making RTM a much more cost effective manufacturing process.

To assist us in this investigation we have developed several techniques which provide quantitative data from which we can measure the quality of the preforms produced, and compare the relative effects of process and material variations. By manipulating this data one can calculate the fabric deformation and tow spreading that occurs as a consequence of forming. The ability to see the distribution and magnitude of these phenomena has provided much insight into the important mechanisms involved in double diaphragm forming. These methods have also allowed us to validate and improve on a simulation of the draping of woven fabric over 3D forms.

### Acknowledgments

We would like to thank Robert Faiz and Nestor Diaz of Dow-United Technologies Composite Products Inc. who funded much of the work described in this paper.

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- [3] Van West, Barry P., "*A Simulation of the Draping and a Model of the Consolidation of Commingled Fabrics*". Center for Composite Materials Technical Report (CCM Report 90-07), Composites Manufacturing Science Laboratory, University of Delaware, 1990.

## STATIC AND FATIGUE TESTING OF FULL-SCALE FUSELAGE PANELS

FABRICATED USING A THERM-X<sup>®</sup> PROCESS

P. 26

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## SUMMARY

Large curved integrally stiffened composite panels representative of aircraft fuselage structure were fabricated using a Therm-X<sup>®</sup> process, an alternative concept to conventional two-sided hard tooling and contour vacuum bagging. Panels subsequently were tested under pure shear loading in both static and fatigue regimes to assess the adequacy of the manufacturing process, the effectiveness of damage tolerant design features cocured with the structure, and the accuracy of finite element and closed-form predictions of postbuckling capability and failure load. Test results indicated the process yielded panels of high quality and increased damage tolerance through suppression of common failure modes such as skin-stiffener separation and frame-stiffener corner failure. Finite element analyses generally produced good predictions of postbuckled shape, and a global-local modelling technique yielded failure load predictions that were within 7% of the experimental mean.

## INTRODUCTION

The manufacture of large composite airframe fuselage structure is greatly facilitated whenever preimpregnated skin, longeron, and frame layups are cocured. A degree of inherent damage tolerance also may be built into the cocured structure by using design concepts which, by their very nature, suppress fundamental postbuckled composite panel failure modes. Conventional manufacturing requirements for cocuring curved stiffened panels mandate the use of precise two-sided hard tooling along with intricate contoured vacuum bagging to ensure high quality consolidation while minimizing defects such as ply wrinkling, fiber bridging, or internal voids. Simplification of both tooling and vacuum bagging requirements would further enhance the cost effectiveness of cocuring.

A manufacturing process using the silicon-based powder polymer Therm-X<sup>®</sup> as a pressure transfer medium was shown to reduce hard tooling and simplify vacuum bagging procedures for cocured skin-hat

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\* Therm-X is a product of the DOW Co.

\*\* Currently with UNITED TECHNOLOGIES-Pratt and Whitney GEB

stiffener and skin-inverted T stiffener building block test specimens [1]. The capability of the same Therm-X process to produce large cocured integrally stiffened fuselage panels with comparable integrity to that noted in [1] was evaluated during this investigation. Specifically, panels were tested under pure shear loading in both static and fatigue environments.

The postbuckled behavior of stiffened composite panels has been investigated by a number of other authors. Prediction of the initial buckling load of stiffened panels, which is often several times less than the ultimate failure load, is usually done through closed-form analyses with various uniform boundary conditions and assumed mode shapes. Due to the complexity of the postbuckled problem formulation, e.g., geometric nonlinearities with intermediate boundary conditions (neither simply supported nor fully clamped), finite element investigations commonly are used instead. References [2-4], for example, used geometrically nonlinear finite element formulations to study the problem of stiffened composite panels loaded in compression. The results cited by these authors indicate generally good correlation between predicted and observed load-deflection behavior but not as good correlation of internal loads, moments, and strains throughout the postbuckled state. Lack of correlation was attributed to initial manufacturing imperfections which were not considered in the numerical studies. The postbuckled behavior of a simply curved multibay stiffened aluminum panel loaded in pure shear was investigated by Jarlas [5]. The physical dimensions of each bay were similar to the composite bays considered in this study. Although the initial buckling load was accurately predicted, convergence difficulties precluded an assessment of overall panel strength while postbuckled. A comprehensive test and analysis program aimed at quantifying postbuckled behavior of curved integrally stiffened panels under combined shear and compression loads was carried out by Ogonowski and Sanger [6]. Approximate closed-form predictions of initial buckling loads and postbuckled load distribution were done using a characteristic panel bay under combined loading. Edge support conditions were either simply supported or clamped on all four sides. In general, all closed-form predictions of postbuckled strength served as upper bounds on experimental results. However, an assessment of local stress states which would initiate failure was not available in closed-form.

In this investigation, prediction of the static failure load of the postbuckled composite panels was done using both closed form methods and finite element analyses. The closed-form method used a modification of the classic diagonal tension analysis developed by Kuhn [7] which accounted for the orthotropic nature of the composite panel [8]. Geometrically nonlinear finite element analyses were used to provide additional information regarding the local states of stress at failure. Using output of a global finite element model, refined local finite element models were built of the vicinity of observed failure locations to investigate local stress states throughout the postbuckled panel.

An investigation of the behavior of one postbuckled panel under cyclic loads was performed. Constant amplitude loading was applied

where the load intensity produced prebuckled and postbuckled skins for the minimum and maximum loads, respectively. The significance of damage growth monitored during the test is discussed.

### AUTOCLAVE THERM-X® PROCESS

Two-sided hard tooling, the conventional concept for cocuring, serves the dual function of properly positioning prepreg layups and evenly distributing autoclave pressure to the composite during the cure cycle. Vacuum bags must conform closely to the surfaces where hard tooling cannot contact the prepreg in order to properly transmit autoclave pressure. Errors in tool positioning can produce undesirable results such as stiffener misalignment, skew, and twist. Inadequate vacuum bag contouring can produce fiber bridging and poor compaction of radius features. Relaxation of tooling and bagging requirements would greatly promote the use of cocure strategy as a viable and economical means of composite manufacture.

A cocure manufacturing concept which takes advantage of the flow characteristics of the silicon-based polymer Therm-X and thereby reduces tooling and bagging requirements has been demonstrated by the authors [1]. Under ambient pressure the Therm-X medium is in the form of a fine powder. When subjected to autoclave pressure, however, the powder exhibits flow characteristics similar to liquid media. In this pressurized flowable state the polymer transmits quasi-hydrostatic pressure to the prepreg that is equal to the applied autoclave pressure. Upon venting to atmospheric pressure, Therm-X reverts back to its powdery state thereby permitting easy tool breakout and laminate removal. There appears to be no limit to the number of times a quantity of Therm-X may be reused in the manner described above.

The advantages of the autoclave cocuring process which uses Therm-X are two-fold. First, only one-sided hard tooling is needed to position the prepreg since hydrostatic pressure exerted by the medium ensures even pressure distribution. Second, only the Therm-X containment vessel (discussed below), and not the entire contour of the laminate, must be vacuum bagged because the magnitude of the pressure within the contained volume of Therm-X depends only on the magnitude of externally applied autoclave pressure.

Fabrication of the curved integrally stiffened panels used in this study was done as outlined in the series of illustrations shown in Figure 1. Note the use of one-sided hard tooling for the frame (inverted T) stiffeners as well as the simplicity of the final vacuum bagging procedure.

### STIFFENED PANEL DESIGN

In order to generate test and analysis information that would be most useful to the industry in general, the test articles were designed to be representative of fixed wing fuselage or rotary wing tailcone structure; see Figure 2. The panel measured 30 inches per edge with



a radius of curvature equal to 40 inches. The skin in the panel bays was made of woven graphite/epoxy (Gr/Ep) oriented 45 degrees to the hat stiffeners and measured 0.030" thick. The frame layup was a symmetric combination of woven and tape forms of Gr/Ep which measured 0.078" thick. Hat stiffener webs were 0.015" thick and made of woven Gr/Ep. Hat stiffener caps were made from a combination of woven and tape Gr/Ep and ranged from 27 to 35 mils thick. In an undamaged condition, the panel was designed to carry an ultimate load of 250 lb/in shear flow per edge. A damage tolerance static strength knockdown of 50% was assumed. To satisfy the damage tolerance ultimate load criterion, the panel should exceed 500 lb/in shear flow at failure.

The objectives of the damage tolerance design features incorporated into the panel were to suppress skin-stiffener separation, a common failure mode of postbuckled panels, and maintain frame-longeron load transfer in the postbuckled state. Stiffener-skin separation which precipitated overall loss of panel stiffness was the most common failure mode observed in Reference [6]. The interface between the skin and stiffener was most often the weakest link in the postbuckled structure because the longerons and frames were cocured directly on top of the skin thereby promoting a free edge induced interlaminar tension stress field. The stiffener-skin separation problem has been investigated by various authors [9,10]. A design alternative which suppresses the separation failure mode is shown in Figure 3 and will henceforth be referred to as the embedded flange design concept. Covering stiffener flange free edges with one ply of skin was expected to reduce the propensity for separation by effectively suppressing the interlaminar tension stress field. Additionally, cocured shear ties between frames and longerons were used in the specimen design, see also Figure 3. These load transfer mechanisms were believed to increase the structural integrity of the panel by ensuring load path continuity in the postbuckled state.

#### TEST DESIGN

A "picture frame" fixture was used to introduce pure shear loading into the specimens. The fixture consisted of four equal length steel I-beam sections pinned at each end to form a square frame enclosing the test article; see Figure 4. Aluminum brackets fastened to the webs of each I-beam were used to restrain the specimens during test. Application of loads along the diagonal as shown in Figure 4 produced nearly pure edgewise shear on the specimen. Loads were applied using an MTS 810 testing machine. Detailed finite element models, with the specimen and fixture modelled explicitly, confirmed that very nearly pure shear was introduced to the panel's two central bays. The fatigue investigation was conducted using the same fixture and MTS unit. Constant amplitude fatigue loads equal to two-thirds of the damage tolerance ultimate requirement, 330 lb/in edge shear, were applied at a frequency of 0.3 to 0.6 Hz and an R-ratio of 0.1. It was felt this load level would ensure some damage growth during the cyclic test. Damage growth was monitored on a decade schedule throughout the test, i.e. at 1, 10, 100, 1000, etc, cycles. Hand-held ultrasonic pulse-echo equipment as well as close visual

inspections served to monitor the progression of damage. Quasi-static strain gage surveys were also performed on the decade schedule to quantify stiffness loss as a function of cycles.

#### STATIC TEST: RESULTS AND ANALYSIS

Three specimens were used for static testing. Each specimen was installed in the picture frame shear fixture, Figure 4, and tested under increasing quasi-static load until catastrophic failure was observed. The panel failed by separating into two large sections. Each panel evidenced initial buckling at about 96 lb/in applied load. The experimental average failure load was 580 lb/in edge shear which corresponds to a postbuckling ratio (PBR) of 6. Each individual test result was in excess of the 500 lb/in damage tolerance ultimate requirement. A summary of initial buckling loads, catastrophic failure loads, and the postbuckling ratio for all specimens is provided in Table 1.

Prior to reaching the initial buckling load, the deflection at the center of the panel was in the direction of the normal to the outer surface (convex side) of the panel. Once the applied load exceeded the initial buckling load a reversal in center panel deflection was observed. From this point onward center deflections were directed along the normal to the inner surface of the panel. While in a postbuckled configuration each panel bay exhibited the classical diagonal buckled pattern, shown in Figure 5 for a load level of 283 lb/in.

Close monitoring of damage generation and growth during testing of panel #3 was representative of all panels. At 487 lb/in shear flow (PBR=5), a 5 inch crack appeared in the web of one outer hat stiffener and grew to 7 inches as load was increased to 554 lb/in. At that load a 4 inch crack appeared in the skin of one bay along a buckle, emanating from a stiffener intersection and producing a change in buckled pattern. A specimen under 554 lb/in shear flow is shown in Figure 6. Final failure occurred at 611 lb/in applied load and is believed to have been precipitated by the cracks in the hat stiffener webs which propagated through the specimen as shown in Figure 7.

Closed-form prediction of static failure loads for the curved stiffened specimens was done using methodology documented in [8]. For the panels in this study, analytically calculated diagonal tension strains at the average failure load of 580 lb/in were 3200 microstrain at the panel bay's center. Strain gage results from the three static tests at hand were reduced using the membrane strain recommendation [6] wherein back-to-back three-element strain rosettes are used to determine midplane laminate strain. Average membrane strain thus determined was 3450 microstrain, a 7.2% difference from the closed-form prediction.

Although the closed-form analysis yielded good predictions of diagonal tension strain at failure, the method is incapable of providing information regarding failure initiation in the hat

stiffener webs. A detailed stress analysis is required to obtain information sufficient to perform a local strength evaluation. A finite element model of the specimen and fixture shown in Figure 8 was run using the geometrically nonlinear solution sequence SOL 66 in MSC/NASTRAN. The iterative solution scheme was controlled through an applied displacement increment technique. At the average failure load of 580 lb/in, the analysis was stopped and all nodal displacements and rotations were output. These finite element results will be referred to as the global solution in the discussions which follow.

As noted above, failure of the specimens was believed to have originated in the webs of the hat stiffeners. Observed local buckles in these webs were thought to have precipitated catastrophic failure. In the global finite element model of the entire specimen the total height of the web was modelled with only one element; see Figure 9. The admissible displacement field of the web, linear between nodes, could not capture the local buckles experimentally observed and as a result could not yield accurate local stress values. For this reason, local modelling efforts focused on the most highly loaded webs, denoted 1 and 3 in Figure 9.

The local finite element model of the entirety of the web of hat stiffener 1 is shown in Figure 10. The height of the web in the local model was modelled using six elements which should allow local buckles to be analytically captured. Boundary conditions around the perimeter of the model were applied by means of specified nodal displacements and rotations at nodes of the local model with exact correspondence to nodes of the global model. These corresponding nodes are circled in the figure. Linear interpolation of all boundary conditions was used for local model boundary nodes in between the nodes with correspondence. The total displacements and rotations from the global model which were associated with the average failure load of 580 lb/in were applied to the boundary of the local model in forty equal increments. The geometrically nonlinear solution scheme used for the global model was used for the local analysis.

At the boundary conditions associated with 580 lb/in edge load, the deflected shape obtained using the local model is shown in Figure 11. The twisting undulations of the web from the leftmost to the rightmost edge shown in this figure were noted during static tests.

The Hoffman criterion [11] was used to assess failure in each finite element. The form of the criterion is defined through the Hoffman Failure Number (HFN):

$$HFN = 1 - (S_{11}^2/X_t X_c) - (S_{22}^2/Y_t Y_c) - (S_{11} S_{22}/X_t X_c) - (X_c - X_t) S_{11}/X_t X_c - (Y_c - Y_t) S_{22}/Y_t Y_c - (S_{12}^2/T^2) \quad (1)$$

when  $HFN > 0$ , no failure of ply,  
when  $HFN \leq 0$ , failure of ply,

where  $S_{11}$  = calculated stress in fiber (or warp) direction,  
 $S_{22}$  = calculated stress in transverse (or fill) direction,  
 $S_{12}$  = calculated in-plane shear stress,  
 $X_t, X_c$  = tension, compression strength in fiber direction,  
 $Y_t, Y_c$  = tension, compression strength transverse direction,  
 $T$  = in-plane shear strength

The criterion allows unequal values of tension and compression strength in the material directions.

Classical laminated plate theory was used to calculate material coordinate system stresses for Eq. (1). Transverse shear loads calculated in the local model were noted to be small and judged not to contribute to failure. Hoffman Failure Numbers for mean strength allowables are presented graphically in Figure 12. The lowest margin lies between contour E and 0.0, the exact value being 0.072. Assuming stresses to scale linearly with load, an acceptable approximation at this point of the postbuckled analysis, the percent error between the analysis and average test results is 7.2%.

A photograph of the location corresponding to the smallest margins in Figure 12 is shown in Figure 13. The photograph was taken from panel #3 after failure. The prediction in Figure 12 is for failure at the top of the web while the observed crack was nearer the bottom. This discrepancy is probably due to either slight boundary overconstraint along the top edge of the model which acts as a modest "load sink" or locally reduced material strength.

The two damage tolerance design features, embedded stiffener flanges and cocured shear ties, performed effectively in the postbuckled regime. As expected, stiffener-skin separation failure modes were suppressed by the embedded flange design. Cocured shear ties remained intact even upon catastrophic failure and as a result were judged to be an effective means of maintaining load transfer between intersecting stiffeners.

#### FATIGUE TEST: RESULTS AND ANALYSIS

One constant amplitude cyclic load fatigue test was performed. It was decided the specimen should produce some damage growth in order to investigate the fatigue capacity of the panel. The maximum fatigue load was selected to be two-thirds of the damage tolerance static ultimate requirement, 330 lb/in edge shear (PBR=3.5). Loss of panel stiffness was monitored through the six strain gage rosettes shown in Figure 14. Substantial life at this maximum load level would demonstrate damage tolerance of the panel under the fatigue environment.

An illustration of the buckled shape of the panel during the first loading cycle is shown in Figure 15. Note the hat stiffeners acted as buckled waveform breakers across which buckling patterns were not continuous. Initial buckling of the panel occurred in the neighborhood of 96 lb/in edge shear, similar to the static tests.

The extension of several visible cracks, denoted A, B, C, and D, during the first 10,000 cycles is highlighted in Figure 16. Based on experience gained during static testing these cracks mainly provided relief of local stress concentrations due to the picture-frame shear loading configuration and did not influence the fatigue life of the panel. The fact that progress of these cracks was arrested for a long time prior to final failure lent validity to the argument. Inspections of the entire panel according to the decade schedule yielded no indications of nonvisible damage.

The test was continued until further damage was noted, see locations E and F in Figure 16. The extent of these delaminations was quantified using a pulse echo ultrasonic technique at the cyclic intervals shown. Once again, these delaminations were judged to relieve local stress concentrations due to the loading configuration and, therefore did not adversely affect the total life of the part. No visible or nonvisible damage in addition to that shown in Figure 16 was identified.

The first significant failure occurred at 69,200 cycles. This failure initiated in both webs of one of the outer hat stiffeners as shown in Figure 17. The cracks were easily visible with the unaided eye and were located approximately halfway between the root and top of the web along the stiffener axis. Extension of the cracks to the sizes shown occurred in a single cycle. Ultrasonic inspection found no new nonvisible damage. The buckled shape of the panel after the first significant failure is presented in Figure 18. Note that while the two undamaged hat stiffeners continued to function as panel breakers, the failed stiffener did not.

Immediately following failure of the hat stiffener, the panel was statically tested to a load of 385 lb/in edge shear, two-thirds of the average ultimate load of 580 lb/in. This test established limit load capability of the damaged panel and served to certify the panel up to that load level and equivalent service flight hours. A second hat stiffener failed at 387 lb/in edge shear, 100% of the test requirement. The failure initiated in the webs of the central hat stiffener at the frame-stiffener intersection corner and grew unstably to the dimensions shown in Figure 19. Nonvisible delamination areas also shown in Figure 19 at the flanges of the two damaged stiffeners were found using pulse echo techniques. All previously existing cracks, which were theorized to be stress relief cracks only and not life limiting cracks, did not extend during this test.

The buckled shape of the panel after the second stiffener failure is shown in Figure 20. Note the central hat stiffener is no longer completely effective as a waveform breaker but the intact stiffener remains effective.

Further cycling of the panel to 200,000 cycles was started immediately following the latter static test. Additional nonvisible damage resulting from this cycling is shown in Figure 21. No new visible cracks initiated during this interval.

As shown in Figure 22, rosettes 3-6 demonstrated significant increases of up to 80% measured shear strain after the test to 385 lb/in. The jumps in strain were interpreted to represent a significant redistribution of load through the specimen which occurred as a result of damage in the two hat stiffeners. At 200,000 cycles the fact that strain readings for rosettes 1 and 2 were 20 to 30% of the values during the first cycle suggests very little load passes through the central bays of the specimen. Rather, the primary load path was around the perimeter of the specimen as evidenced by the sharp increases in strain noted in rosettes 3-6 after the test. Since this new load path no longer worked the gage section of the specimen the fatigue test was discontinued.

A final damage summary for the fatigue test article is provided for convenience in Figure 23.

### CONCLUSIONS

Major conclusions resulting from the present investigation are summarized below.

- i) The autoclave Therm-X process used in this investigation can be used to cocure large integrally stiffened curved panels effectively and with a high degree of quality.
- ii) Both closed-form analytical and numerical finite element predictions of static test failure load were good to within 7% of the experimental average. Only the global-local finite element approach could predict the actual location of failure.
- iii) Damage tolerance design ultimate load requirement, 500 lb/in edge shear, was surpassed during all static tests. Average static failure load was 580 lb/in edge shear.
- iv) Constant amplitude fatigue tests with maximum load equal to two-thirds of the damage tolerance ultimate load requirement, i.e. 330 lb/in edge shear, yielded a fatigue life of 69,200 cycles. Residual strength tests demonstrated a second hat stiffener failure at 100% of the target load intensity (two-thirds of the experimental mean).
- v) Damage tolerance design features cocured with the structure, i.e. embedded stiffener flanges and frame-longeron intersection shear ties, were effective by suppressing undesirable failure modes at skin-stiffener interfaces and stiffener intersections during both static and fatigue testing.

### ACKNOWLEDGMENT

The authors would like to acknowledge the concerted efforts of the Sikorsky Aircraft Materials and Processes Test Lab staff during all phases of this program. In particular, the contributions of Mr. Dave Tuttle to the success of the test program deserve special mention.

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TABLE 1: STATIC TEST RESULTS SUMMARY

| SPECIMEN NO | FINAL FAILURE<br>LOAD (LB/IN) | POSTBUCKLING<br>RATIO* |
|-------------|-------------------------------|------------------------|
| 1           | 566                           | 5.90                   |
| 2           | 563                           | 5.86                   |
| 3           | 611                           | 6.36                   |
| Average     | 580                           | 6.04                   |

\* Based on initial buckling load of 96 lb/in edge shear



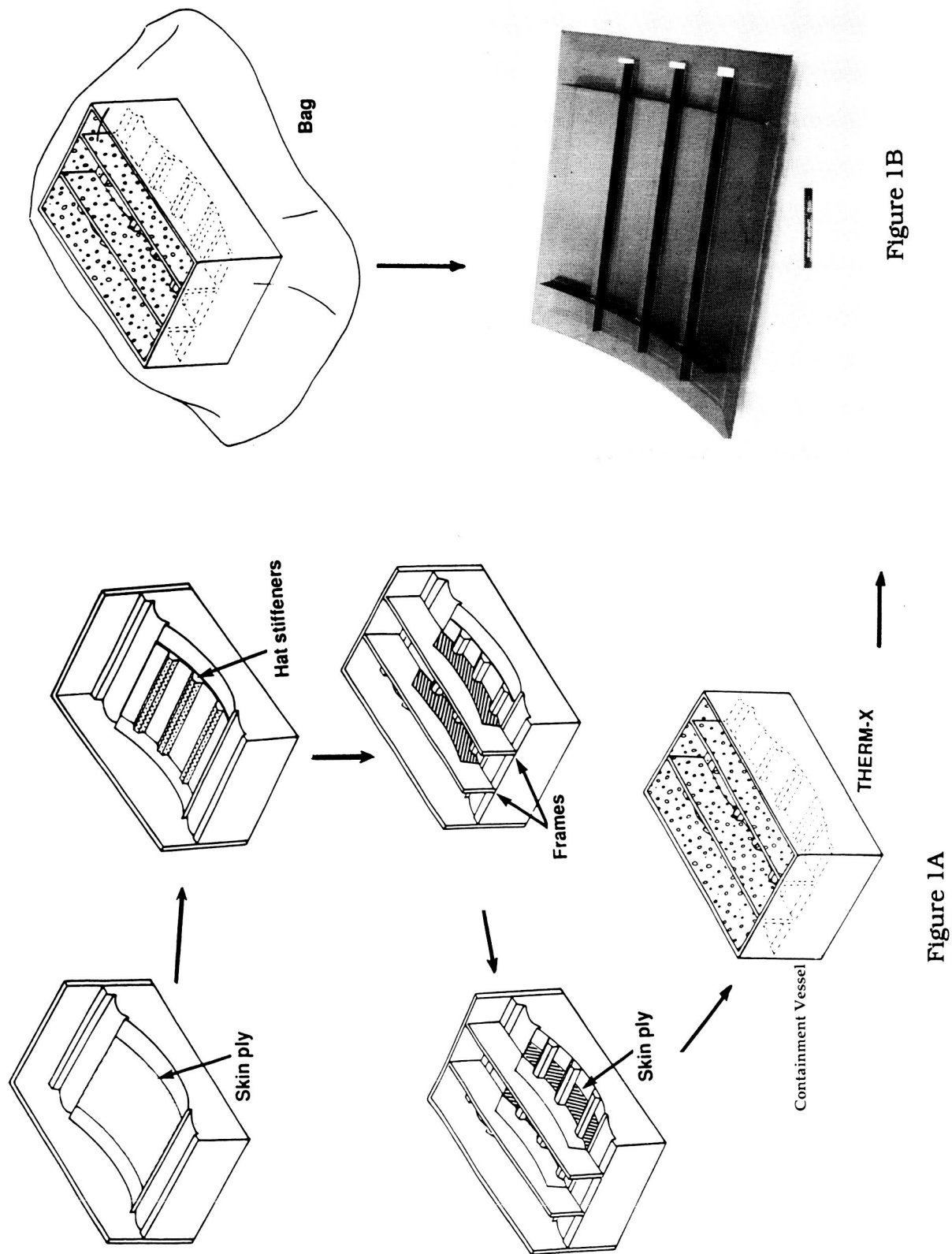
## List of Figures

### Static and Fatigue Testing of Full Scale Fuselage Panels Fabricated Using a Therm-X Process

by

Albert J. DiNicola, Christos Kassapoglou, and Jack C. Chou  
UNITED TECHNOLOGIES - Sikorsky Aircraft Division

- Figure 1: Fabrication Sequence for Curved Stiffened Panels
- Figure 2: Representative Fuselage and Tailcone Panels
- Figure 3: Shear-Tie and Embedded Flange Design Concepts
- Figure 4: The Picture Frame Shear Fixture
- Figure 5: Diagonal Buckled Pattern,  $q = 283$  lb/in Edge Shear
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- Figure 19: Damage At 387 lb/in During Residual Strength Test
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- Figure 22: Shear Strain vs. Cycles
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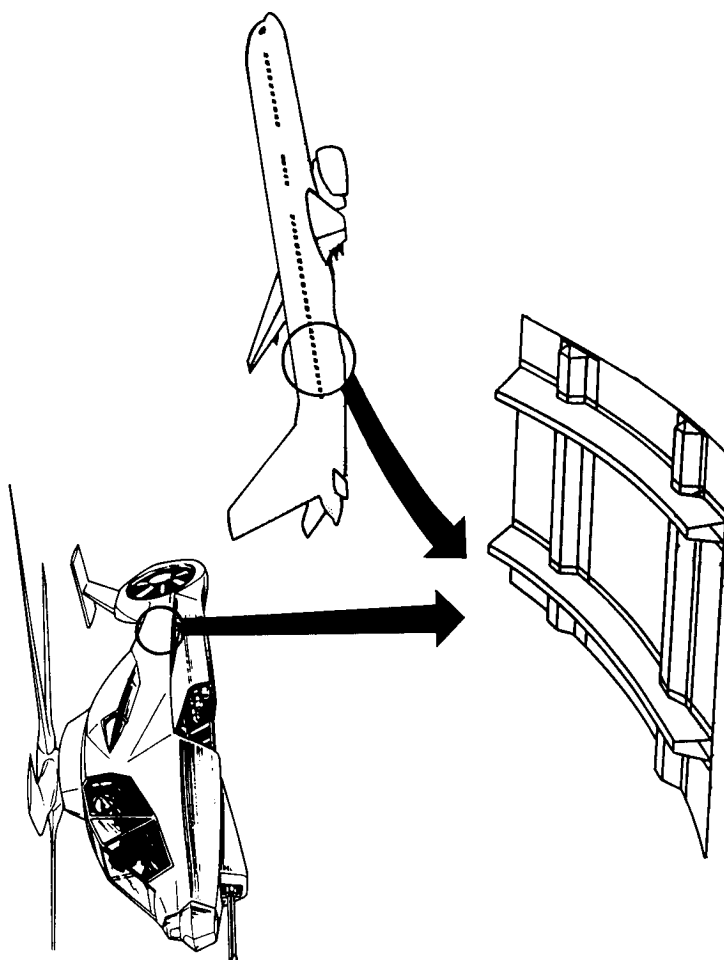


Figure 2

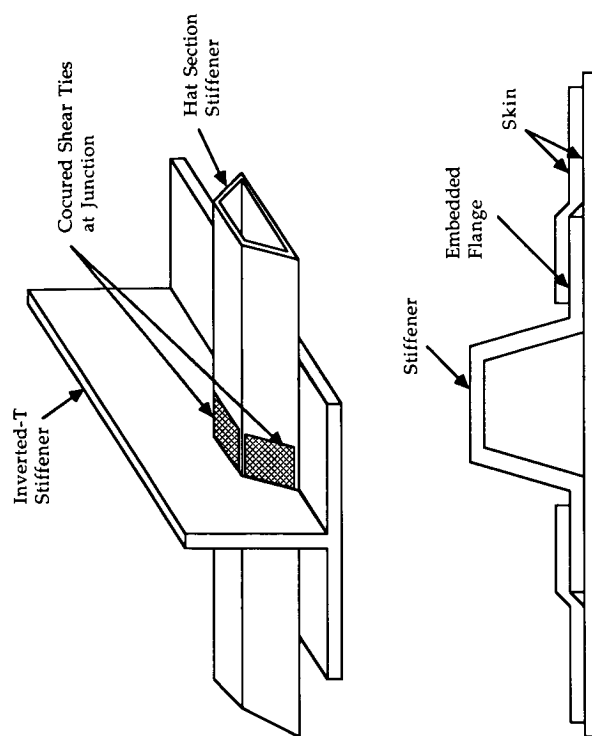


Figure 3



Figure 4



Figure 5

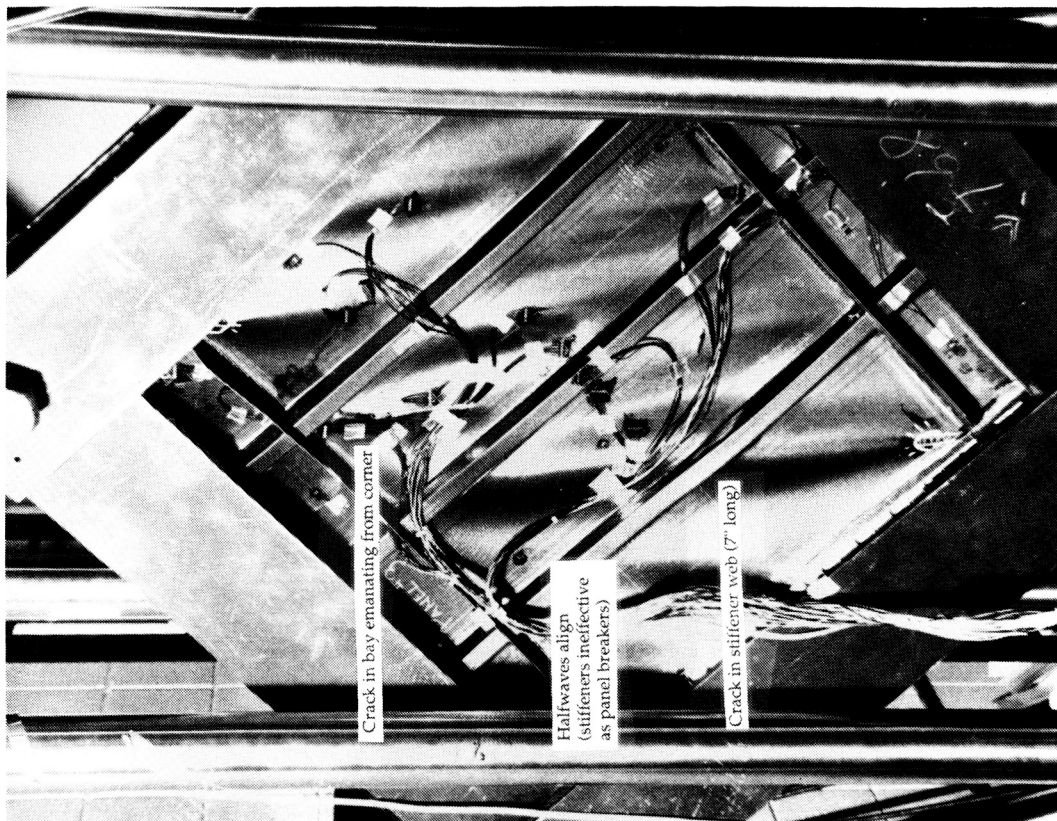


Figure 6

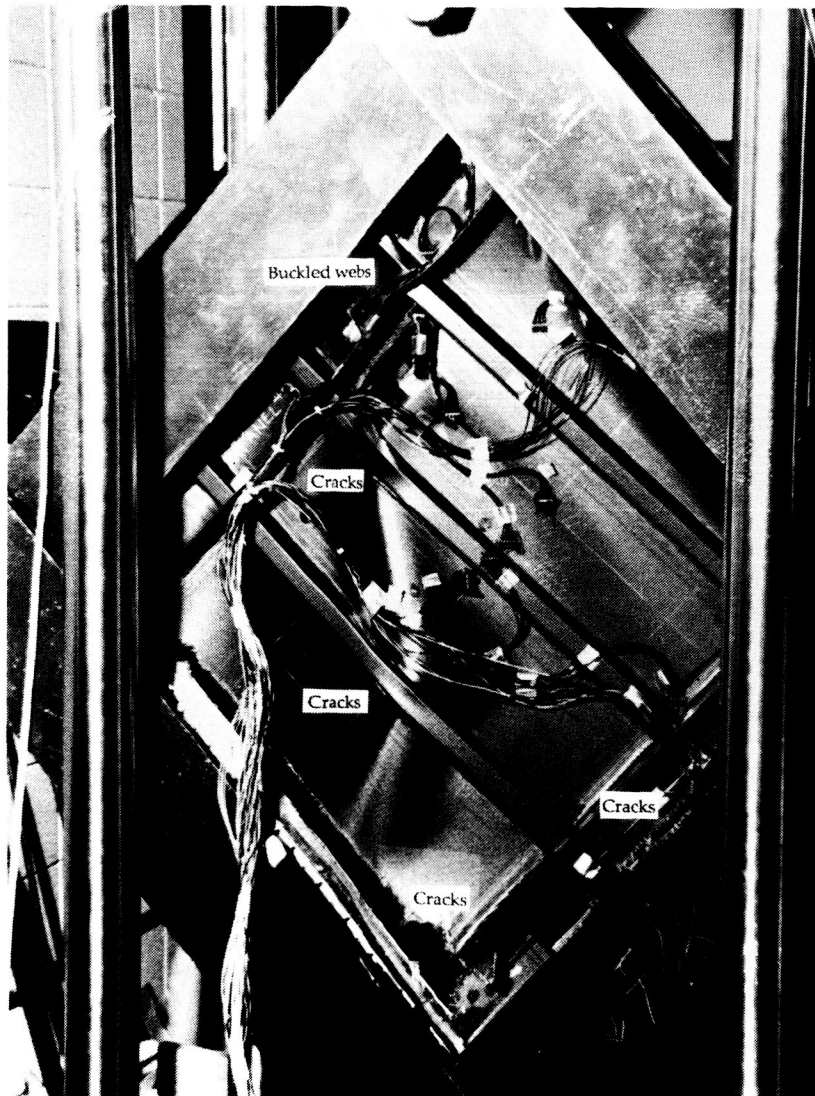


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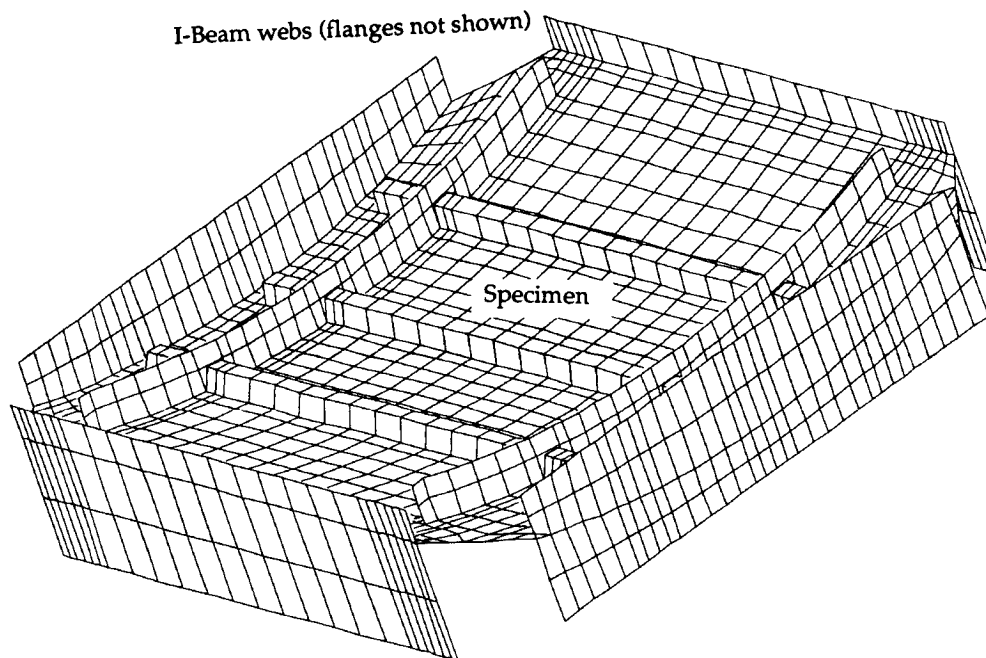


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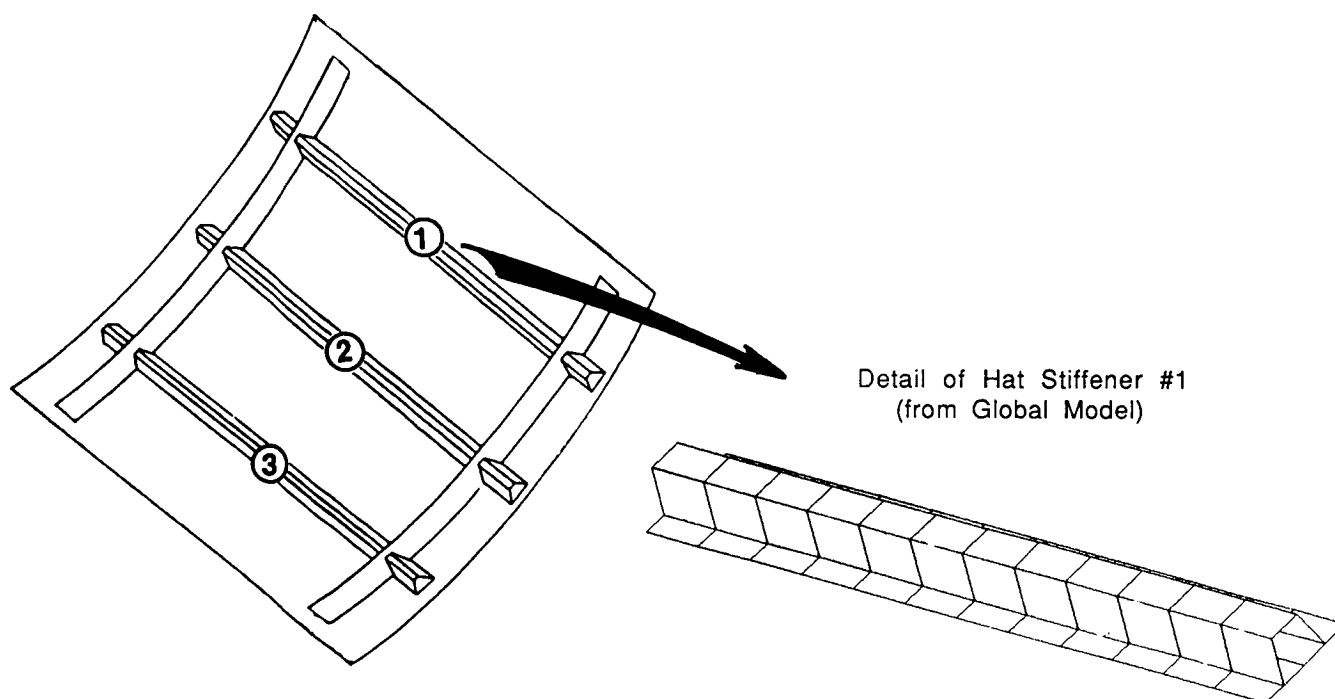


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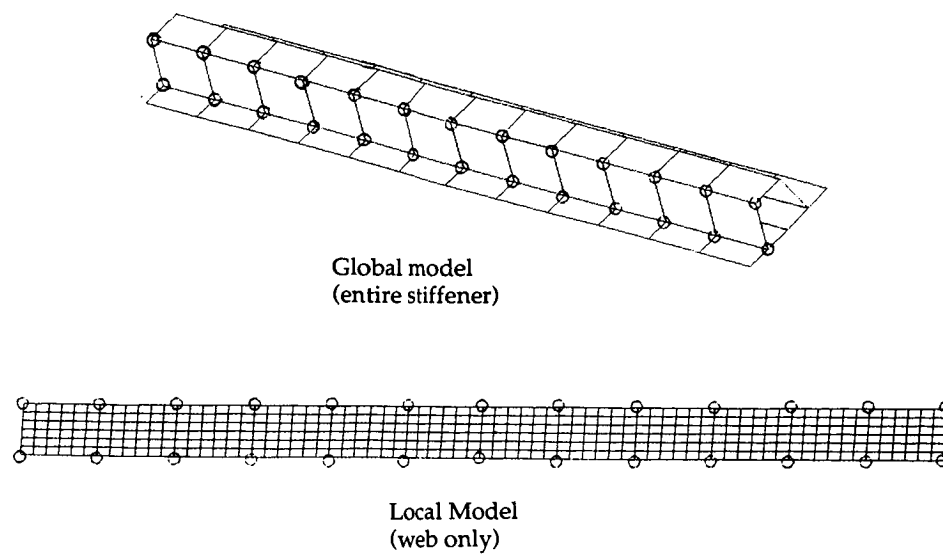


Figure 10

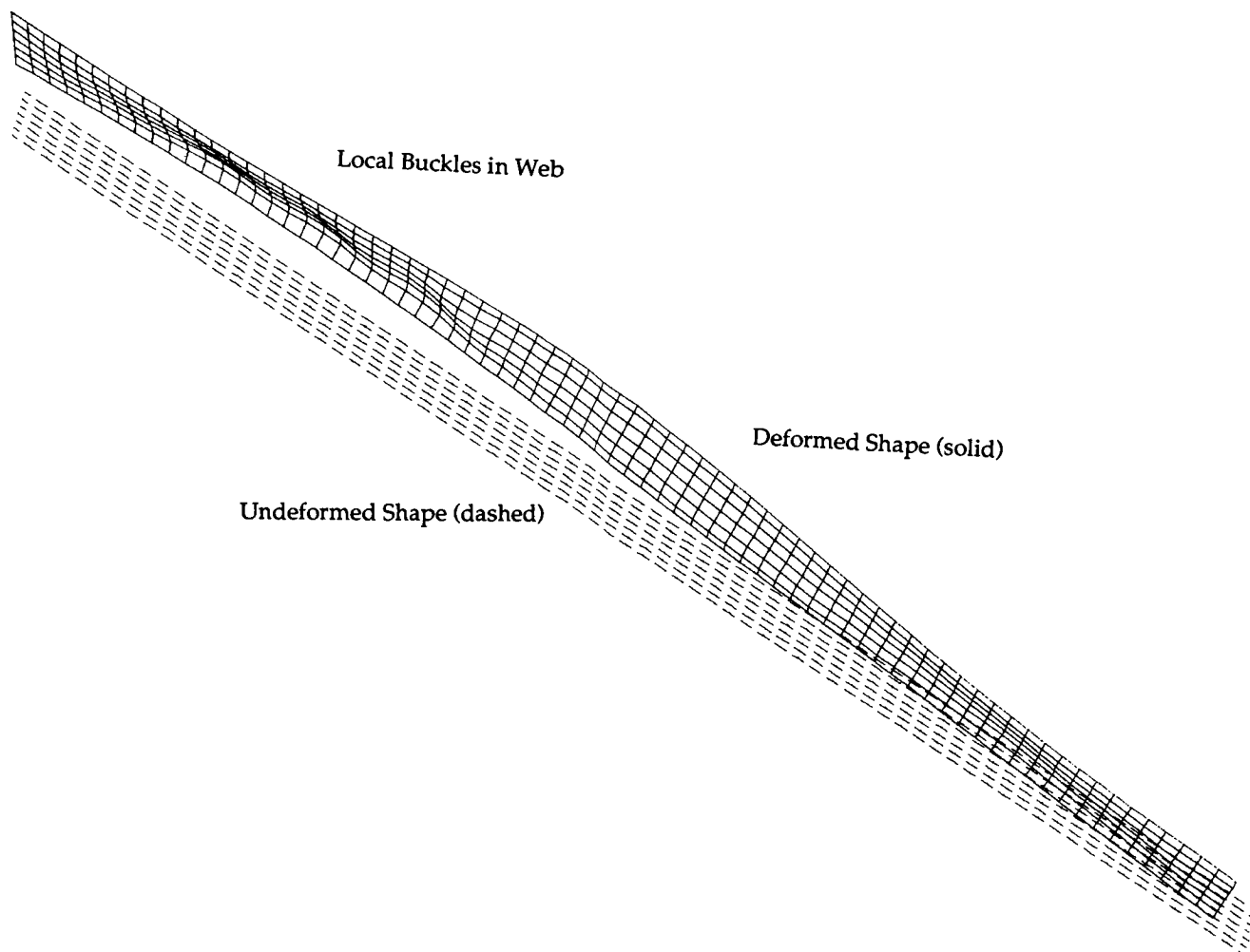
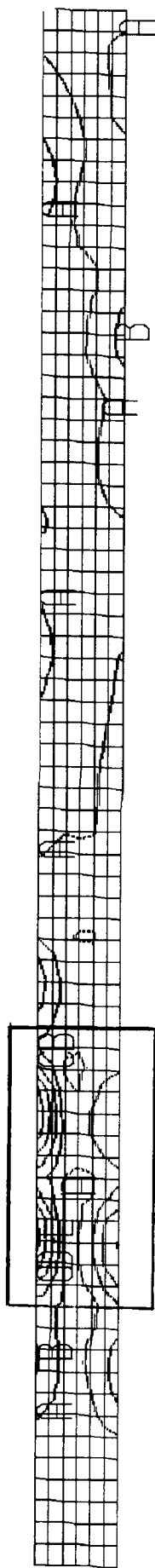
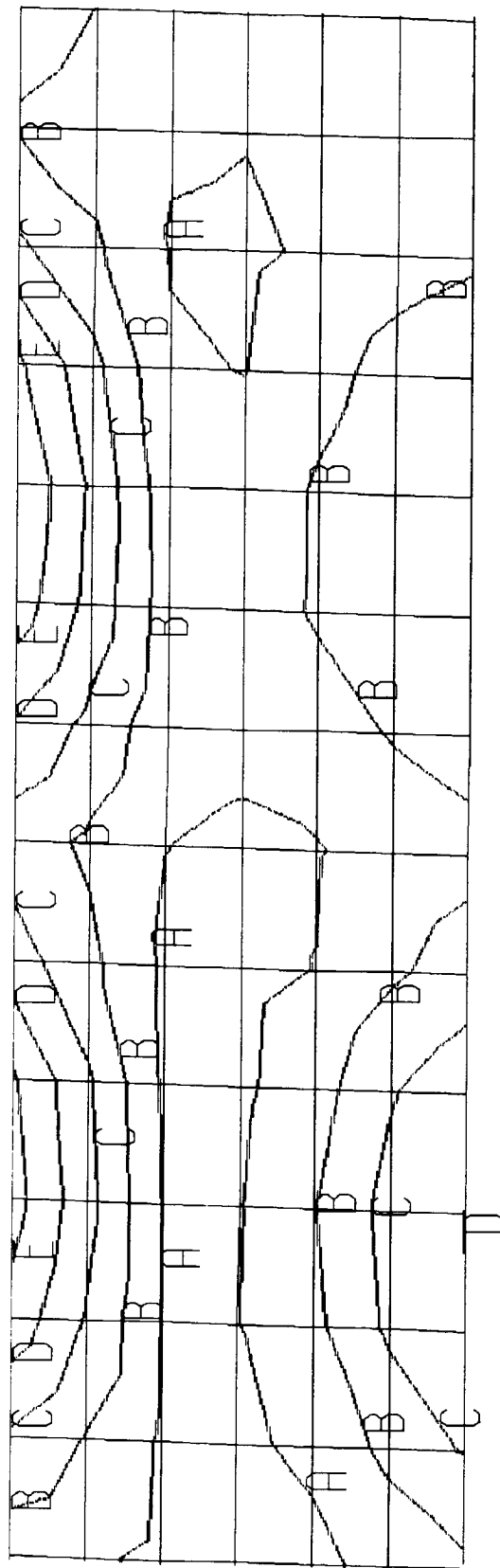


Figure 11





Overall



Detail

| Contour<br>Letter | Hoffman Failure<br>Number (Mean) |
|-------------------|----------------------------------|
| A                 | 0.750                            |
| B                 | 0.583                            |
| C                 | 0.416                            |
| D                 | 0.250                            |
| E                 | 0.082                            |

FIGURE 12

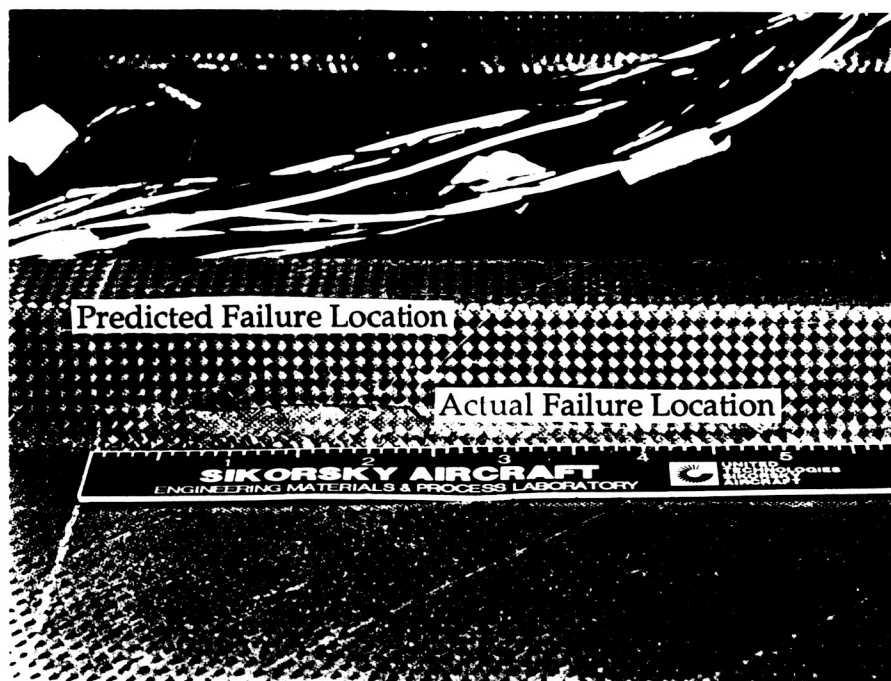


FIGURE 13

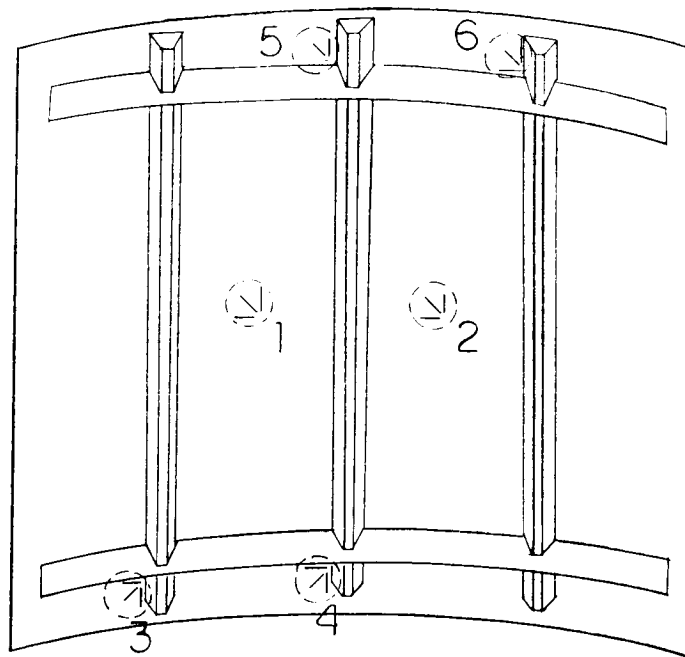


Figure 14

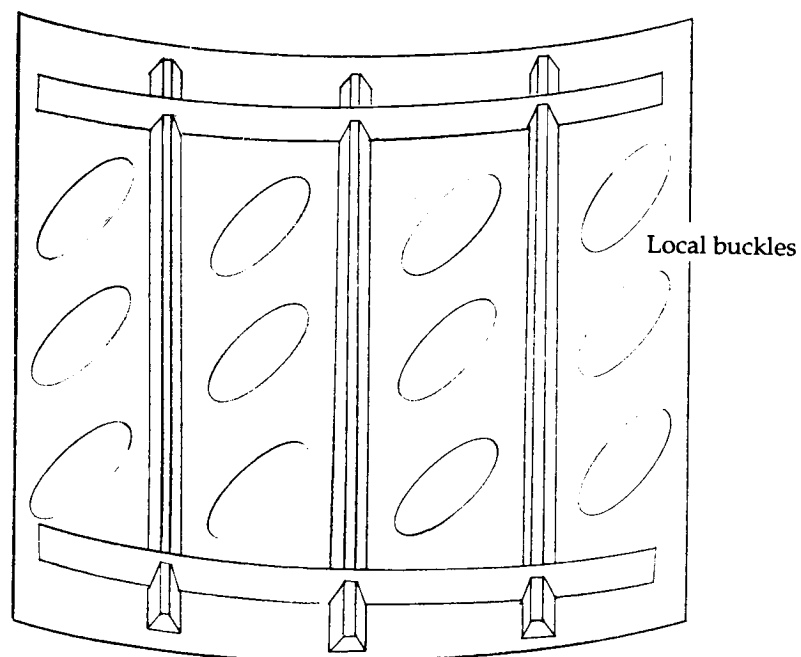


Figure 15

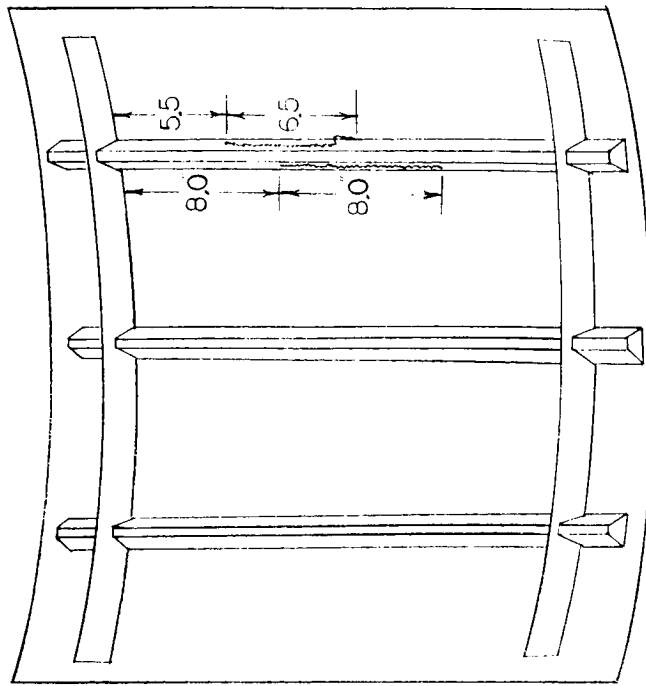


Figure 17

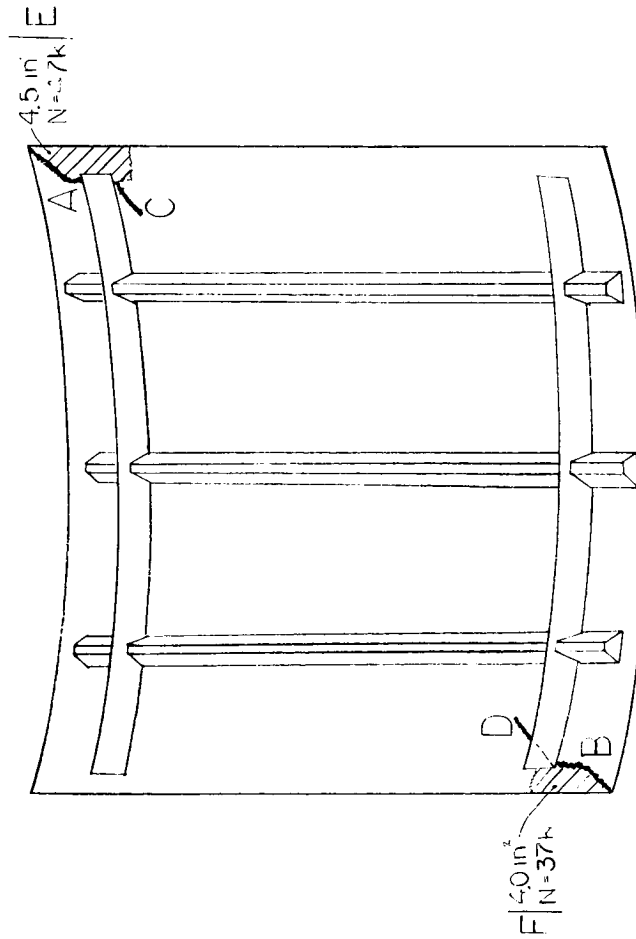


Figure 16

| Crack | Crack Length [in] |           |            |              |               |
|-------|-------------------|-----------|------------|--------------|---------------|
|       | 1 cycle           | 10 cycles | 100 cycles | 1,000 cycles | 10,000 cycles |
| A     | 1.50              | 1.50      | 2.75       | 2.75         | 2.75          |
| B     | 1.50              | 1.50      | 1.50       | 2.50         | 2.50          |
| C     | ----              | 0.75      | 0.75       | 2.00         | 2.00          |
| D     | ----              | ----      | 0.50       | 0.50         | 2.50          |

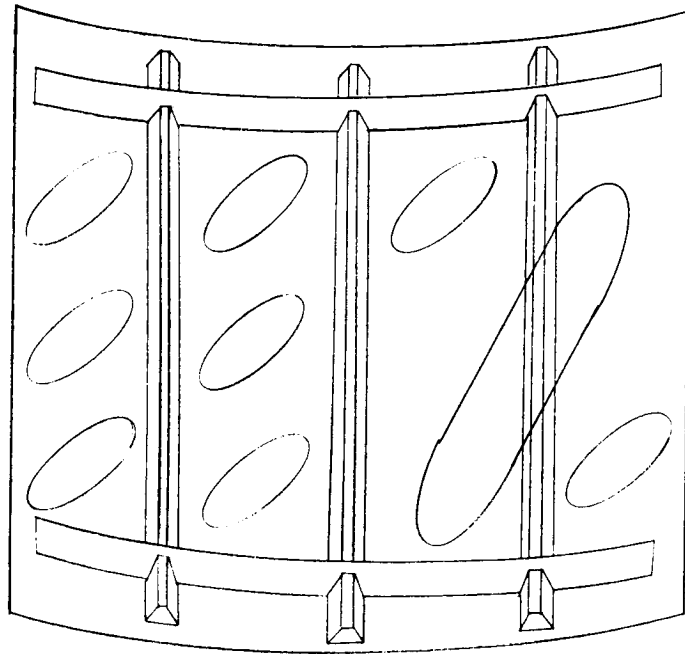


Figure 18

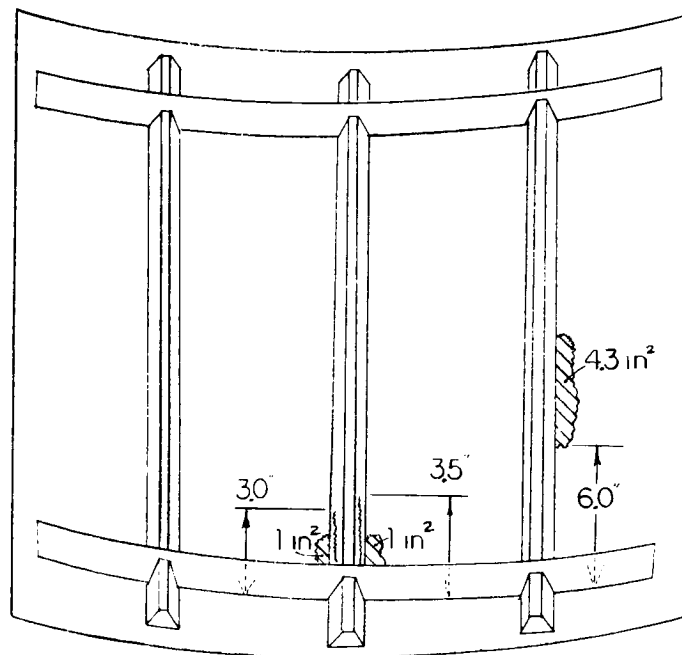


Figure 19

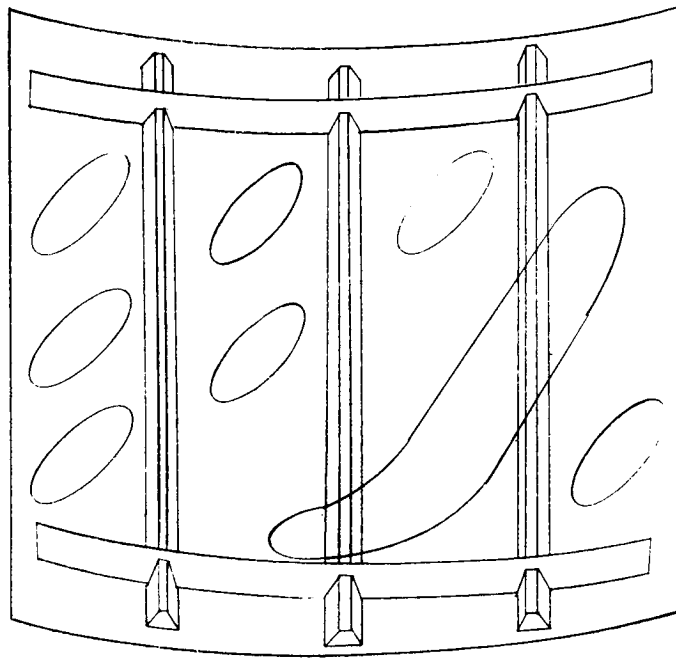


Figure 20

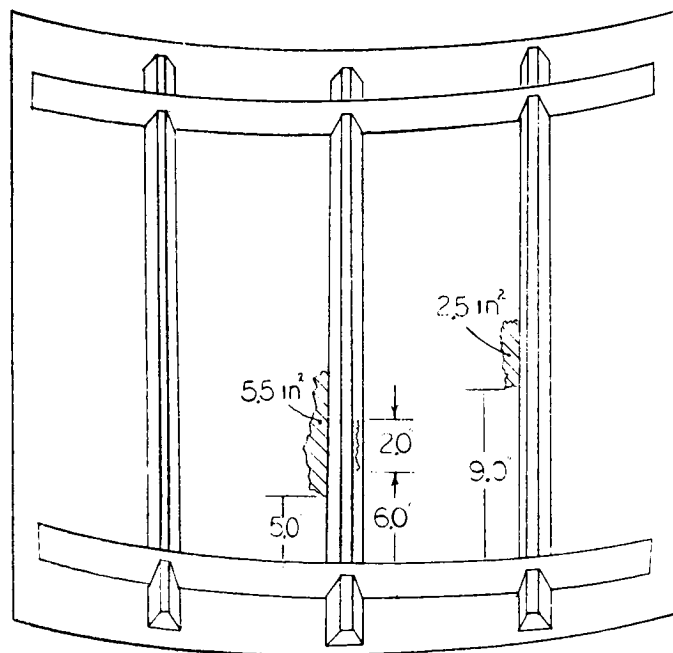


Figure 21

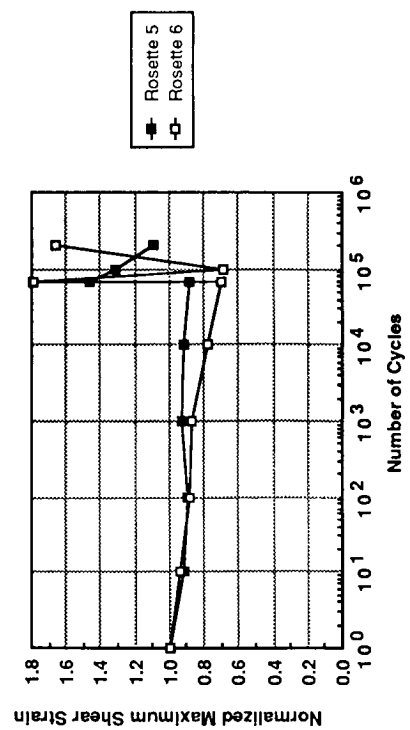
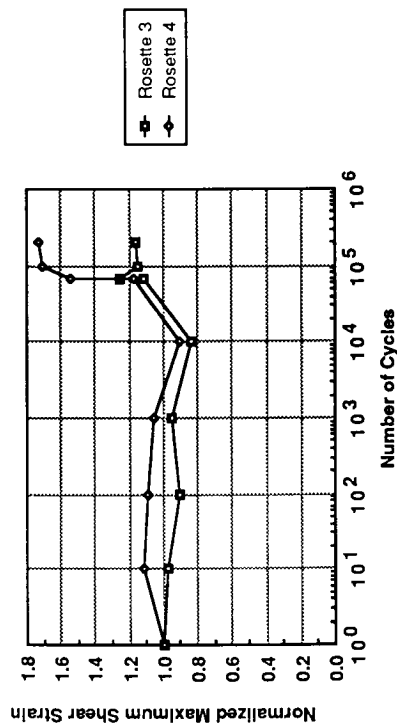
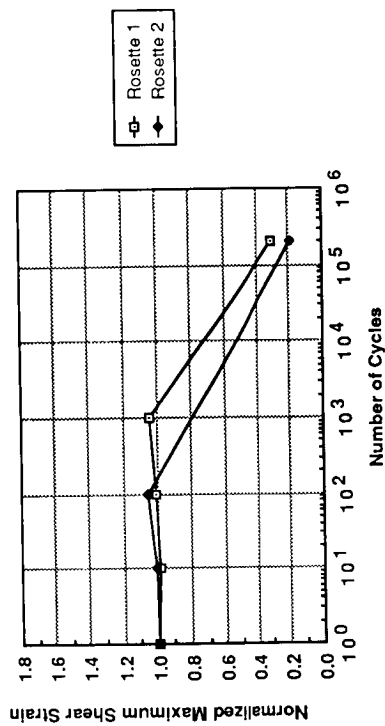
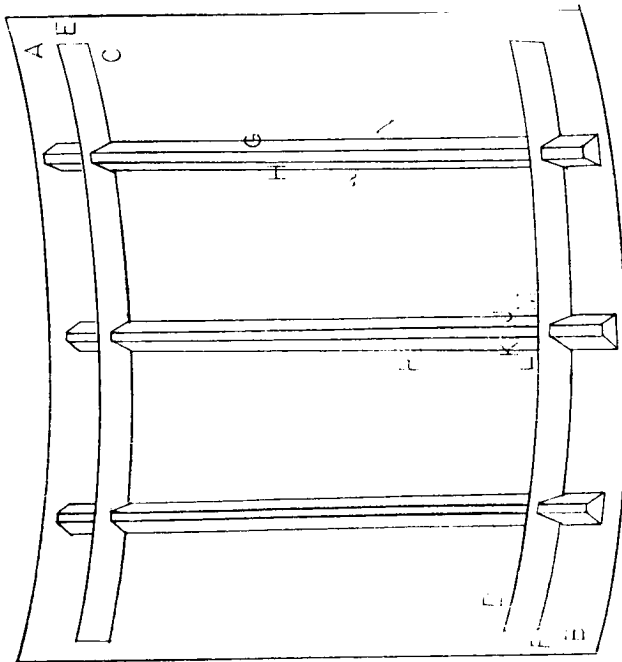


Figure 22



| Damage Site | Type  | Final Size |
|-------------|-------|------------|
| A           | Crack | 2.8 in     |
| B           | Crack | 2.5 in     |
| C           | Crack | 2.0 in     |
| D           | Crack | 2.5 in     |
| E           | Delam | 4.5 in**2  |
| F           | Delam | 4.0 in**2  |
| G           | Crack | 6.5 in     |
| H           | Crack | 8.0 in     |
| J           | Crack | 5.5 in     |
| K           | Crack | 3.0 in     |
| L           | Delam | 1.0 in**2  |
| M           | Delam | 1.0 in**2  |
| N           | Delam | 4.3 in**2  |
| P           | Delam | 5.5 in**2  |
| Q           | Delam | 2.5 in**2  |

Figure 23

**ADVANCED TOW PLACEMENT OF COMPOSITE  
FUSELAGE STRUCTURE**

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**ABSTRACT**

The Hercules NASA ACT program was established to demonstrate and validate the low cost potential of the automated tow placement process for fabrication of aircraft primary structures. The program is currently being conducted as a cooperative program in collaboration with the Boeing ATCAS Program. The Hercules advanced tow placement process has been in development since 1982 and was developed specifically for composite aircraft structures. The second generation machine, now in operation at Hercules, is a production-ready machine that uses a low cost prepreg tow material form to produce structures with laminate properties equivalent to prepreg tape layup.

Current program activities are focused on demonstration of the automated tow placement process for fabrication of subsonic transport aircraft fuselage crown quadrants. We are working with Boeing Commercial Aircraft and Douglas Aircraft during this phase of the program. The Douglas demonstration panel has co-cured skin/stringers, and the Boeing demonstration panel is an intricately bonded part with co-cured skin/stringers and co-bonded frames.

Other aircraft structures that have been evaluated for the automated tow placement process include engine nacelle components, fuselage pressure bulkheads, and fuselage tail cones. Because of the cylindrical shape of these structures, multiple parts can be fabricated on one tow placement tool, thus reducing the cost per pound of the finished part.

**CONFERENCE**

Ninth DoD/NASA/FAA Conference on Fibrous Composites in Structural Design,  
4-7 November, 1991, Lake Tahoe, Nevada.

**HERCULES ACT PROGRAM OBJECTIVE**

Composite materials have demonstrated significant weight savings for aircraft structures with the added advantages of outstanding corrosion and fatigue damage resistance. Despite these advantages, the potential benefits of composite aircraft primary structures have been limited by the high cost of materials, labor intensive manufacturing processes, and inadequate technology in structural mechanics and materials science.

The objective of the Hercules ACT Program is to use an automated seven-axis tow placement machine in development of low cost manufacturing processes for efficient aircraft structural forms. Specifically, Hercules will demonstrate the advanced tow placement process for fabrication of subsonic transport aircraft fuselage structures.



## HERCULES AUTOMATED TOW PLACEMENT

Hercules began the development of tow placement technology for the automated placement and in-process consolidation of ribbonized prepreg tow in 1980. In 1983, our first machine (FPM1) was operational and was used to manufacture flat panels, curved panels, and 360° cross sections, including stiffened and unstiffened skins.

Hercules tow placement process makes use of robotic machine technology to provide an automated fabrication process for high performance composite structures. The process involves the precise automated placement and in-process compaction of ribbonized prepreg tow. Multiple tows are laid down as a band, with band location and angle precisely controlled. Material cut and add features, incorporated into the process, provide high production rate potential, enhance design tailorability, and minimize material scrap.

Hercules has successfully demonstrated the capability to fabricate a wide variety of complex structures using this technology. Aircraft wing components, including ribs and spars, air inlet ducts, and fuselage structures, have been successfully tow placed.

Hercules currently has two operational tow placement machines. FPM1 is a six-axis machine that has the capability to manufacture structures with a 20-ft maximum length and 11-ft maximum swing diameter. Our new production-rated machine (FPM2 shown in Figure 1), which became operational in early 1990, is a seven-axis machine that has the capability to manufacture structures with a 33-ft length and 13-ft swing diameter.

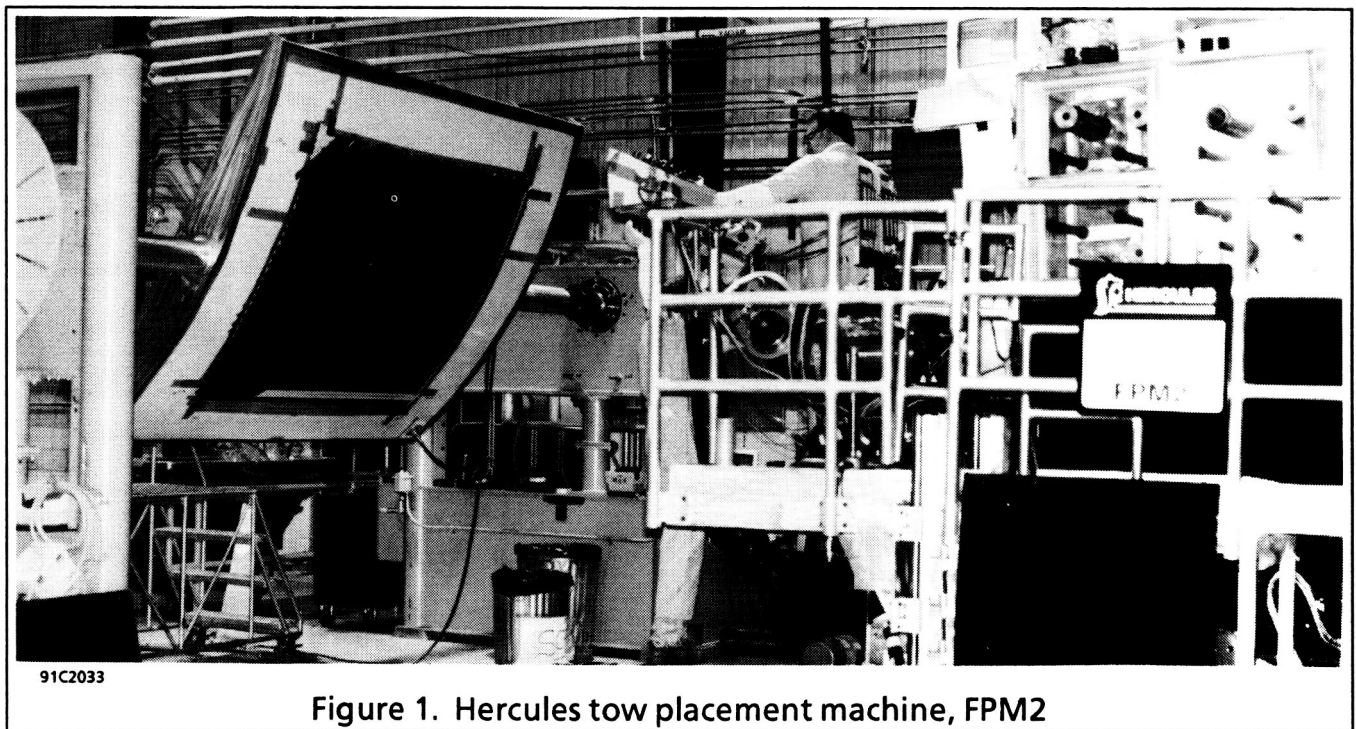
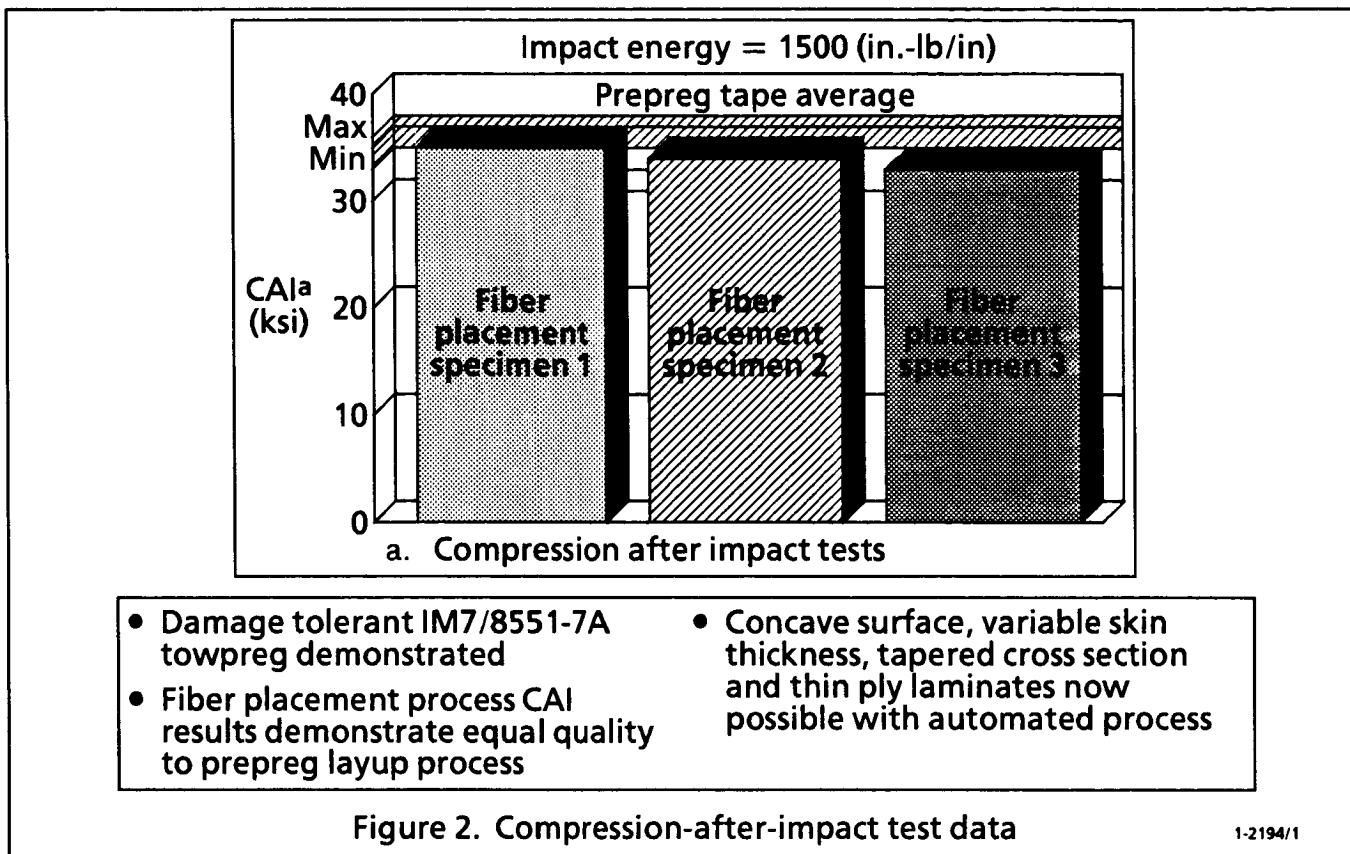


Figure 1. Hercules tow placement machine, FPM2

These machines use a prepreg tow material form that is projected to be approximately 20% lower in cost than prepreg tape. The tow-placed product is also comparable in performance to hand layup prepreg tape parts (Figure 2).

Hercules tow placement technology has continued to mature and improve during the past year. Various improvements have been made to the second generation machine (FPM2). Improvements have been made in our fiber placement delivery heads, machine control systems, and off-line programming. The fourth generation band cut/add (BCA)



fiber placement delivery head was built; this head incorporates the successful features of the past heads plus some new innovative features to make it more production worthy. The key features of the fourth generation BCA head are the 32-tow, 4-1/4-in. bandwidth capability, the easy to remove components for quick servicing, and the synchronization of fiber lay down and band adding. The first Cure-On-The-Fly™ delivery head was also built. The Cure-On-The-Fly™ head is used in an on-line process for delivering thermoset prepreg tow requiring only a bagless oven cure, avoiding an autoclave cure cycle. In addition, the efficiency by which fiber path data are paged and manipulated was improved to decrease the computer time needed to generate control tables for the fiber placement machine. Also, the off-line programming computer is being upgraded. The IBM RS6000 Model 550 computer has been selected to replace the Apollo DN580.

### HERCULES NASA ACT PROGRAM

In early 1990, the Hercules ACT Program was redirected from an isogrid stiffened fuselage structure to a more conventionally stiffened fuselage with hat section and blade stringers. The redirection process took several months to complete, but work was resumed in January 1991. The revised Hercules program was set up as a cooperative program between Hercules and the Boeing ATCAS Program. Hercules will fabricate test panels that are representative in design of crown, keel, and side quadrants of a Boeing Commercial transport aircraft. All panels will be fabricated using the automated tow placement process. In addition to providing designs, Boeing will test all panels fabricated for the Hercules ACT Program (Table 1).

Tow placement of the flat crown panels was completed in July and August 1991. Two 60-in. × 150-in. flat unstiffened 15-ply panels were tow placed and cured in July 1991 (Figure 3). These panels were delivered to Boeing and will be tested for uniaxial damage tolerance. One of the panels was a hybrid of fiberglass/ graphite and the other was an all

| Fuselage Quadrant                                                | Test Article                                        | Undam-<br>aged<br>Elements | Tension<br>With<br>Damage | Shear<br>With<br>Damage | Comp.<br>With<br>Damage | Bi-<br>Tension<br>With<br>Damage | Comp/<br>Shear<br>With<br>Damage |
|------------------------------------------------------------------|-----------------------------------------------------|----------------------------|---------------------------|-------------------------|-------------------------|----------------------------------|----------------------------------|
| Crown                                                            | Flat, unstiffened skin panels, 60 in. X 150 in.     |                            | 2                         |                         |                         |                                  |                                  |
| Crown                                                            | Flat, stiffened panels, 63 in. X 150 in.            |                            | 1                         |                         |                         |                                  |                                  |
| Crown                                                            | Curved, stiffened panels, 65 in. X 72 in.           |                            |                           |                         |                         | 1                                |                                  |
| Keel                                                             | Flat, coupons, 5 in. X 7 in.                        |                            |                           |                         | 24                      |                                  |                                  |
| Keel                                                             | Flat, stiffened panels, 30 in. X 44 in.             |                            |                           |                         | 6                       |                                  |                                  |
| Keel                                                             | Curved, stiffened panels, 30 in. X 44 in.           |                            |                           |                         |                         |                                  | 1                                |
| Window belt                                                      | Tension coupons with thick taper, 12 in. X 12 in.   | 3                          | 3                         |                         |                         |                                  |                                  |
| Window belt                                                      | Curved panel with taper and cutout, 40 in. X 40 in. |                            | 1                         |                         |                         |                                  |                                  |
| Window belt                                                      | Panel with double window frame 40 in. X 40 in.      |                            |                           | 1                       |                         |                                  |                                  |
| Table 1. Test matrix for Boeing/Hercules ACT Program integration |                                                     |                            |                           |                         |                         |                                  |                                  |

graphite panel. The hybrid panel consisted of 25% S2 glass and 75% AS4 fiber. Both the glass and graphite tows were impregnated with Fiberite 938 resin. The all graphite panel was tow placed with AS4 fiber impregnated with Fiberite 938 resin. The panels were cured using a 350°F and 100 psi cure cycle with a hold at 225°F. Bleeder release plies were applied to both sides of the panels to help allow volatiles to escape. Glass rovings were attached along the edge of the panels to also help the volatiles escape. Teflon film was placed over the panels that prevented resin from bleeding out, but still allowed air to escape through the glass rovings. A caul plate was placed over the panels and they were bagged and cured. Both panels looked good. Neither panel had a problem with trapped volatiles.

Tow placement of a 63-in. X 150-in. flat stiffened panel was also completed in July 1991 (Figure 4). The panel has five 16-ply hat stringers co-cured to the 15-ply skin. Both the panel skin and hat stringers are a hybrid material form consisting of 25% S2 glass and 75% AS4 graphite. The glass and graphite tows were impregnated with Fiberite 938 epoxy resin. The hat stringers for this panel are fabricated from a 16-ply panel that was tow placed on a large flat mandrel. The hat stringers will be kitted from this panel and hot drape formed in a machined aluminum forming tool. The cure process for this stiffened panel will use the inside mold line (IML) flex caul that has been used successfully at Hercules on other stiffened panels. The molded four-ply flex caul will be made from a machined REN 550 master model. The hat stringer cure mandrels will be machined metal tools. At the time of this paper, this panel had not been cured, but the skin panel and stringer panel had been bagged and stored in the freezer. We are waiting for delivery of the flex caul model and stringer forming tool.

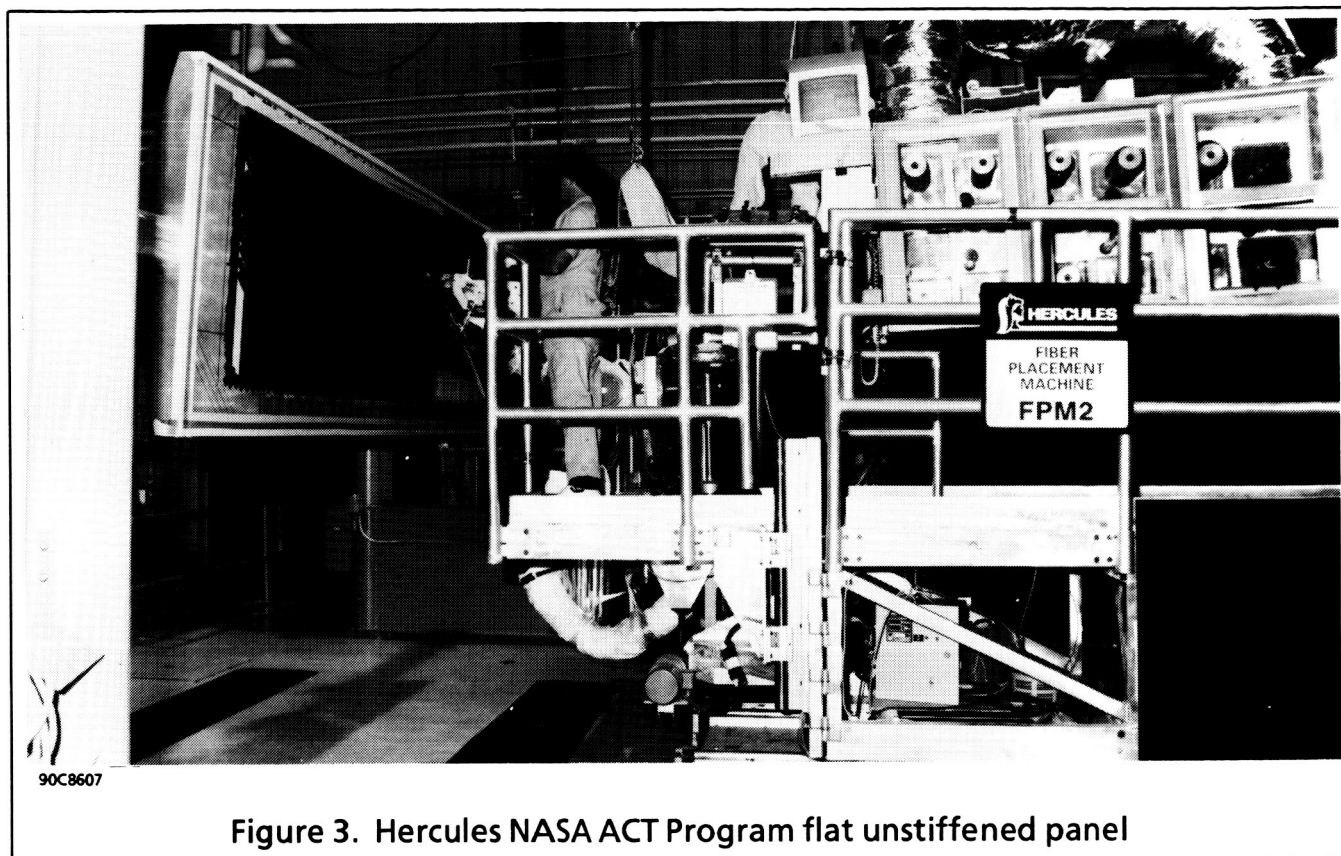


Figure 3. Hercules NASA ACT Program flat unstiffened panel

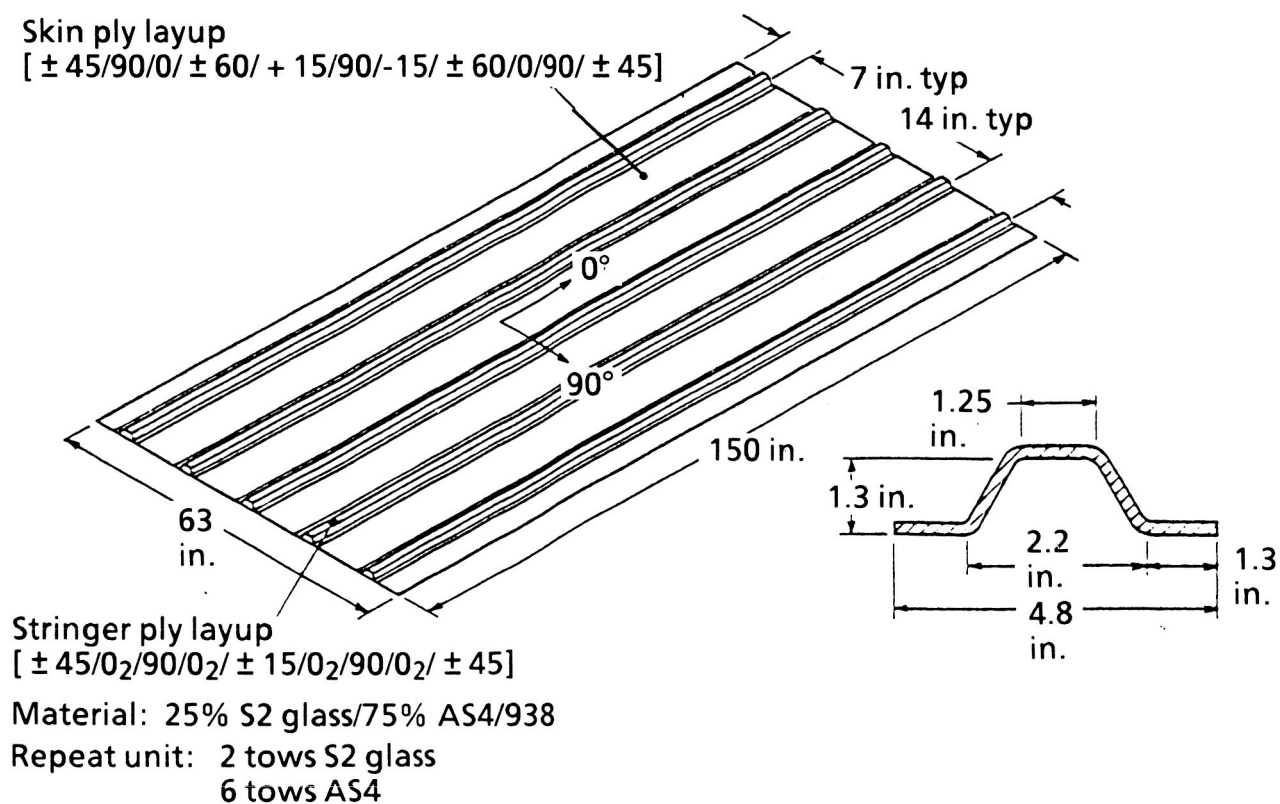


Figure 4. Hercules NASA ACT Program flat stiffened panel

The last item of the Hercules program representative of the fuselage crown section is a 65-in. X 72-in. curved panel with hat stringers. This panel will also have three J frames. The tow-placed skin and stringers will be co-cured and the RTM frames will be co-bonded. This panel will be fabricated in October 1991.

Fabrication of the fuselage keel and window belt structural test panels will be completed in 1992 and 1993.

### HERCULES NASA ACT SUBCONTRACTS

Hercules advanced tow placement was selected for evaluation on other NASA ACT contracts. We currently have subcontracts from the Douglas ICAPS Program and Boeing ATCAS Program (Table 2). The Boeing ATCAS subcontracts are ongoing and include both tow placement of test panels and fabrication of tooling. The Douglas ICAPS subcontract was recently completed and we look forward to working with Douglas in Phase B if they continue their evaluation of tow placement.

|                                    |                                                                                                                                                                                                                                                                  |
|------------------------------------|------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|
| Boeing ATCAS                       | Unstiffened flat panels (5 each)<br>Stiffened flat panels (1 each)<br>3-ft X 5-ft curved stiffened panels (2 each)<br>Tear strap panels (4 each)<br>7-ft X 10-ft curved stiffened panels (2 each)<br>10-ft X 12-in. oval mandrel<br>8-ft X 12-ft Invar cure mold |
| Douglas ICAPS                      | Stiffened skin panels                                                                                                                                                                                                                                            |
| Table 2. Hercules ACT subcontracts |                                                                                                                                                                                                                                                                  |

### BOEING ATCAS SUBCONTRACTS

Several Boeing ATCAS subcontracts are ongoing at this time and two have recently been completed. The following paragraphs describe all Boeing ATCAS subcontracts to Hercules during 1991.

**Flat Unstiffened Panels** Five flat unstiffened coupon panels were tow placed in July 1991 under subcontract from the Boeing ATCAS program. Four of the panels were 40 in. X 132 in. and one panel was 60 in. X 150 in. The four 40-in. X 132-in. panels were for biaxial tension testing and the 60-in. X 150-in. panel was for hoop damage tolerance testing. Two of the biaxial tension panels were tow placed with a hybrid of S2 glass/938 resin and AS4-6K/938 resin. The other two biaxial panels were all graphite using AS4-6K/938 resin. One of the hybrid panels had hybrid material only in the 0°plies of the laminate. The 60-in. X 150-in. hoop damage tolerance panel was an all graphite AS4/938 panel. These panels were delivered to Boeing and test data are included in the Boeing ATCAS conference paper.

**Flat Stiffened Panel** A flat stiffened panel was tow placed in July 1991 under subcontract from the Boeing ATCAS Program. This panel is 63 in. X 150 in. and is stiffened by five hat stringers. The hat stringers are kitted from a large tow-placed panel and then hot drape formed. The skin and stringers will be co-cured using the Hercules IML flex caul and machined metal stringer cure mandrels. At the time of this paper, both the stringer panel and skin panel had been tow placed and stored in the freezer. We are waiting for delivery of tooling items required for cure.

**3-Ft X 5-FT Curved Stiffened Panel** Two 3-ft X 5-ft curved stiffened panels are being fabricated under subcontract from the Boeing ATCAS Program. Both panels have a radius

of 122 in. and are stiffened with three co-cured hat stringers and three co-bonded J frames. One of the panels is a hybrid of S2 glass/938 resin and AS4/938 resin. The other panel is an all graphite panel made with AS4/938. Both skin and stringers are made with tow-placed material. The skins were tow placed on a large 10-ft×12-ft oval mandrel with a radius of 122 in. The stringer panel was tow placed on a large flat mandrel. Both panels will be cured on an 8-ft×12-ft Invar outside mold line (OML) cure mold. The IML side tooling was designed by Boeing and enables the co-cure of skin/stringers and co-bond of the precured frames all in one autoclave cycle. At the time of this paper, the hybrid skin and stringer panels were tow placed and stored in the freezer awaiting delivery of the Invar cure mold. The all graphite panel was scheduled for tow placement in September 1991.

**Tear Strap Panels** We are expecting a subcontract in September for four tear strap panels. Three of the panels will be flat 30-in.×100-in. panels. Two of these panels will have precured tear straps co-bonded to the skin with a layer of film adhesive. The other flat panel will use no film adhesive between tear straps and skin. The fourth tear strap panel will be a 65-in.×72-in. curved skin tow placed on the 122-in. radius mandrel and cured in the Invar cure mold. It will have precured tear straps co-bonded to the skin with a layer of film adhesive. The subcontract for these panels will be finished before the November ACT conference.

**7-Ft×10-Ft Curved Stiffened Panel** A subcontract for two 7-ft×10-ft curved stiffened panels is expected in October. At the time of this paper, final design of these panels had not been defined. They will be scaled up from the 3-ft×5-ft panel discussed in this paper. The tool approach will be the same as that used for the 3-ft×5-ft. panels. An RFQ has not been received at this time, but is expected in September. Fabrication of these panels will be in October and November 1991.

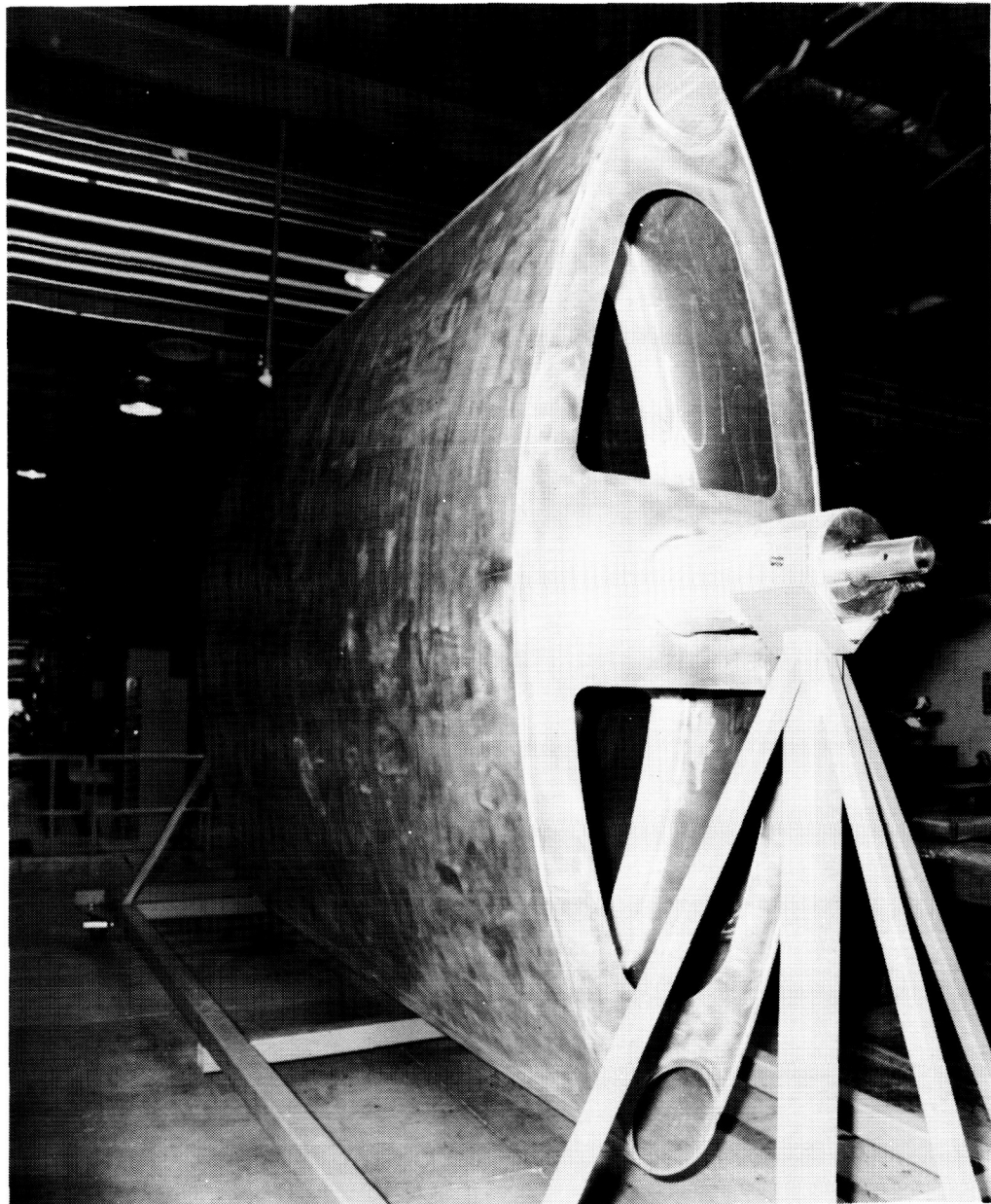
**10-Ft×12-Ft Oval Mandrel** A large 10-ft.×12-ft tow placement mandrel (Figure 5) was fabricated in June 1991 under subcontract from the Boeing ATCAS Program. This mandrel has two sides in an oval configuration and each side has a radius of 122 in. The mandrel was made by rolling two aluminum plates to the 122-in. radius and welding the plates to an aluminum support structure. The mandrel shaft is a machined thick wall aluminum tube that was also welded to the support structure. This tool will be used for tow placement of the ATCAS 3-ft.×5-ft panels and 7-ft×10-in. panels.

**8-Ft×12-Ft Invar Cure Mold** An 8-ft×12-ft Invar cure mold (Figure 6) was fabricated in July and August 1991 under subcontract from the Boeing ATCAS Program. This concave cure mold has a 122-in. radius cure surface. The tool was fabricated by rolling a 0.750-in. thick plate of Invar 36 steel to the required 122-in. radius. Invar ribs (0.250-in. thick) were cut to size with a water jet and welded to the rolled plate. After welding the support structure, the tool was stress relieved and set up in a large three-axis machine for final machining of the 122-in. radius tool surface. Invar was selected for this tool because of coefficient of thermal expansion (CTE) concerns. Invar is a low CTE material. Because we are using this tool to co-bond a cured frame to an uncured skin, we did not want our cure tool to grow much at 350°F as this would cause a poor quality bond between the frame and skin. If co-bonding of the frames had not been a requirement, we would have selected a lower cost material such as aluminum for the cure mold.

## **DOUGLAS ICAPS SUBCONTRACT**

The Hercules subcontract from the Douglas ICAPS program was recently completed. The objective of this contract was the fabrication of tow-placed panels that would be compared with identical panels made with the RTM process at Douglas. The automated tow placement (ATP) panels and RTM panels will be compared for structural performance and cost effectiveness.

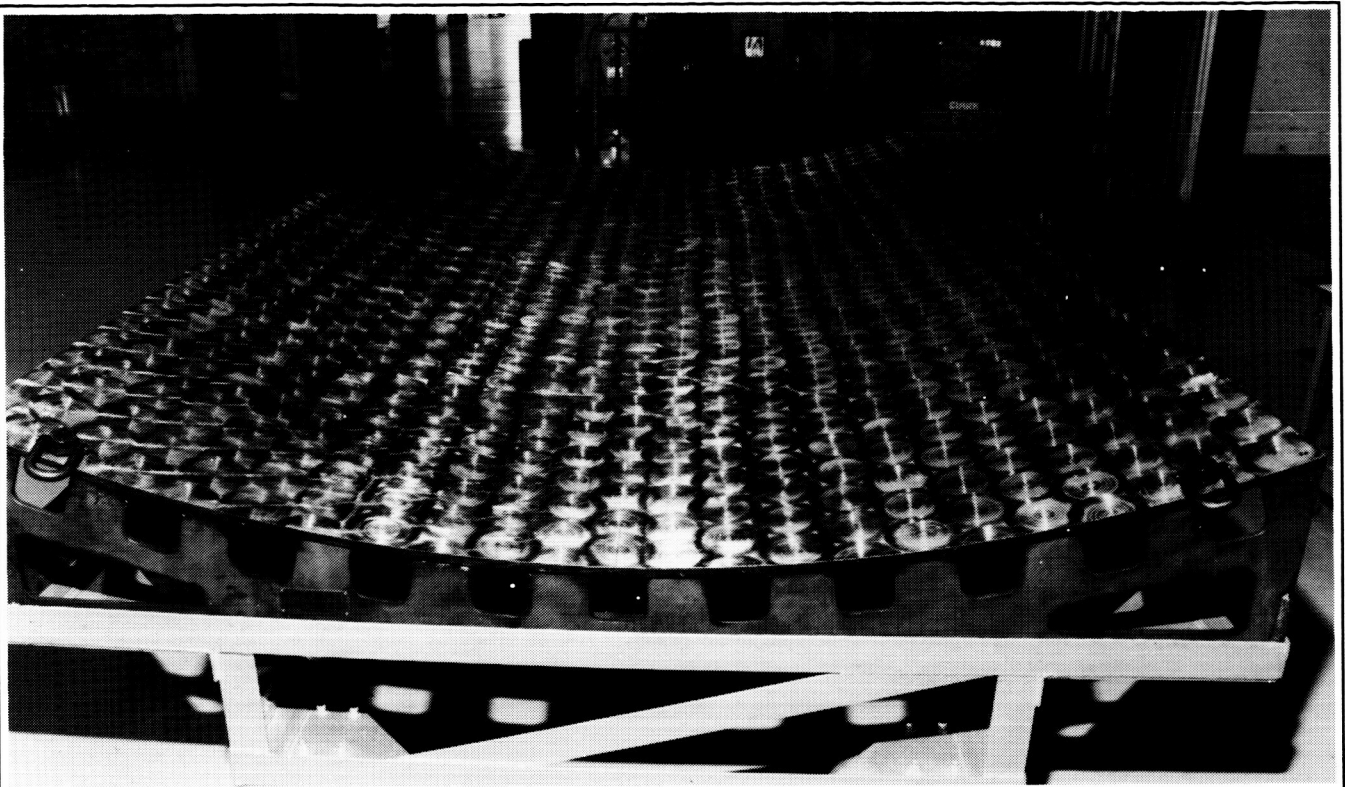




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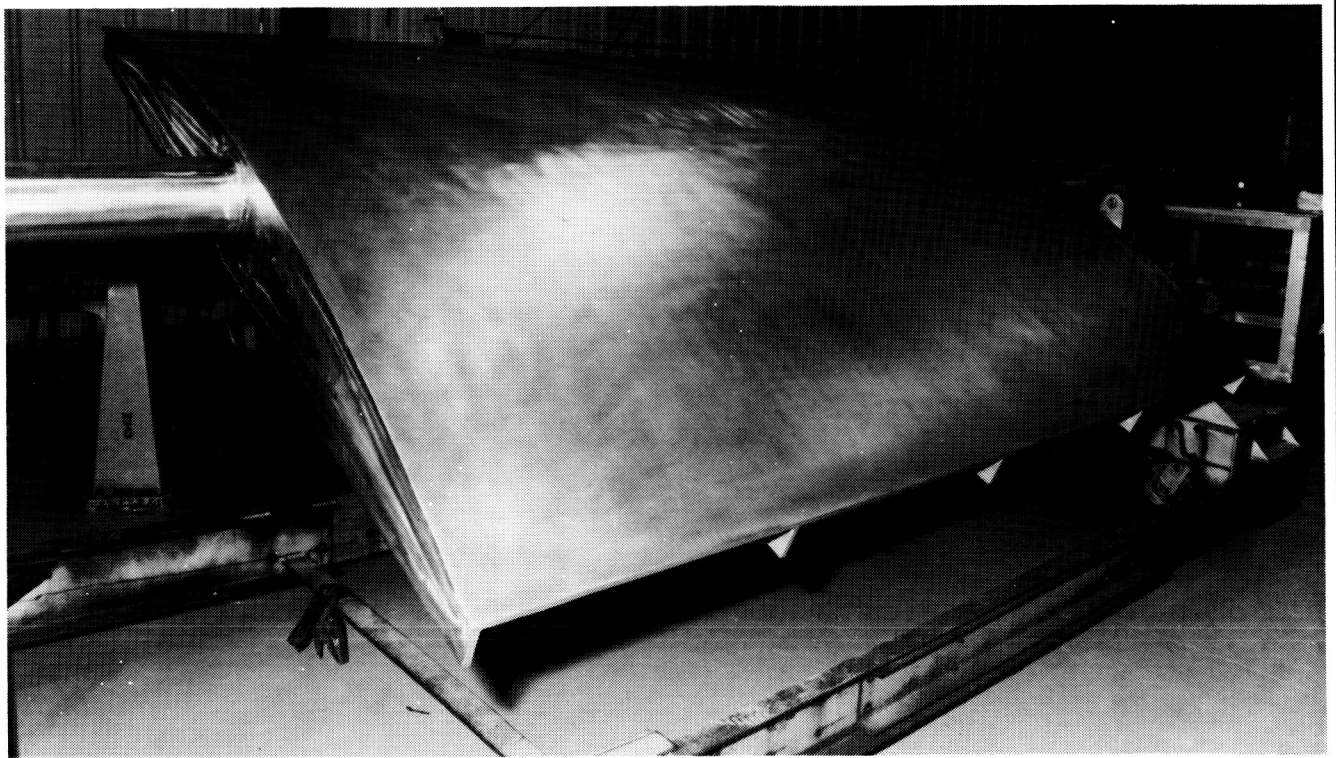
Figure 5. Boeing ATCAS tow placement mandrel

The tooling concept for the ICAPS tow-placed panels was somewhat different than anything previously used at Hercules. Our objective was a low cost, low risk tool concept to achieve a skin to stringer co-cure. Surface smoothness was also a consideration in our tool concept selection. The stiffened test panels simulated aircraft fuselage skin so we wanted the outside mold line (OML) surface to be as smooth as possible. Some other objectives we wanted to achieve with our tool concept were a uniform skin thickness, close tolerance in spacing of the stringers, and net shape of the stringer achieved during the panel cure process. To accomplish these objectives, we used a low cost aluminum mandrel (Figure 7) to tow place the panel skin and transferred the skins to an aluminum OML mold for cure. The OML cure mold achieved the skin smoothness we wanted. The stringer spacing tolerance, uniform skin thickness, and net shape stringers were accomplished by using a molded caul sheet on the IML side of the panel during cure.



91C7427

Figure 6. Boeing ATCAS Invar Cure Mold



91C1539

Figure 7. Douglas ICAPS subcomponent panel tow placement mandrel



We used this tool concept on the flat element panels and the 118-in. radius subcomponent panels. The element panels were 21 in.  $\times$  36 in. and were stiffened with three J-stringers. The subcomponent panels were 48 in.  $\times$  60 in. and were stiffened with six J-stringers. The subcomponent panels also had three shear tee doublers that ran perpendicular to and under the stringers. The shear tee doublers were also co-cured to the panel skin.

The tooling concept was very successful for both the flat element panels and the large 118-in. radius subcomponent panels. The tooling was simple and easy to use and produced excellent quality panels.

The most innovative feature of our tool concept was the use of a four-ply molded flex caul on the IML side of the panel. The flex caul was laid up on a master model machined from monolithic graphite (Figure 8). Detail was machined into the model for stringer cavities and shear tee doublers. The flex caul was laid up on the model using tooling prepreg.

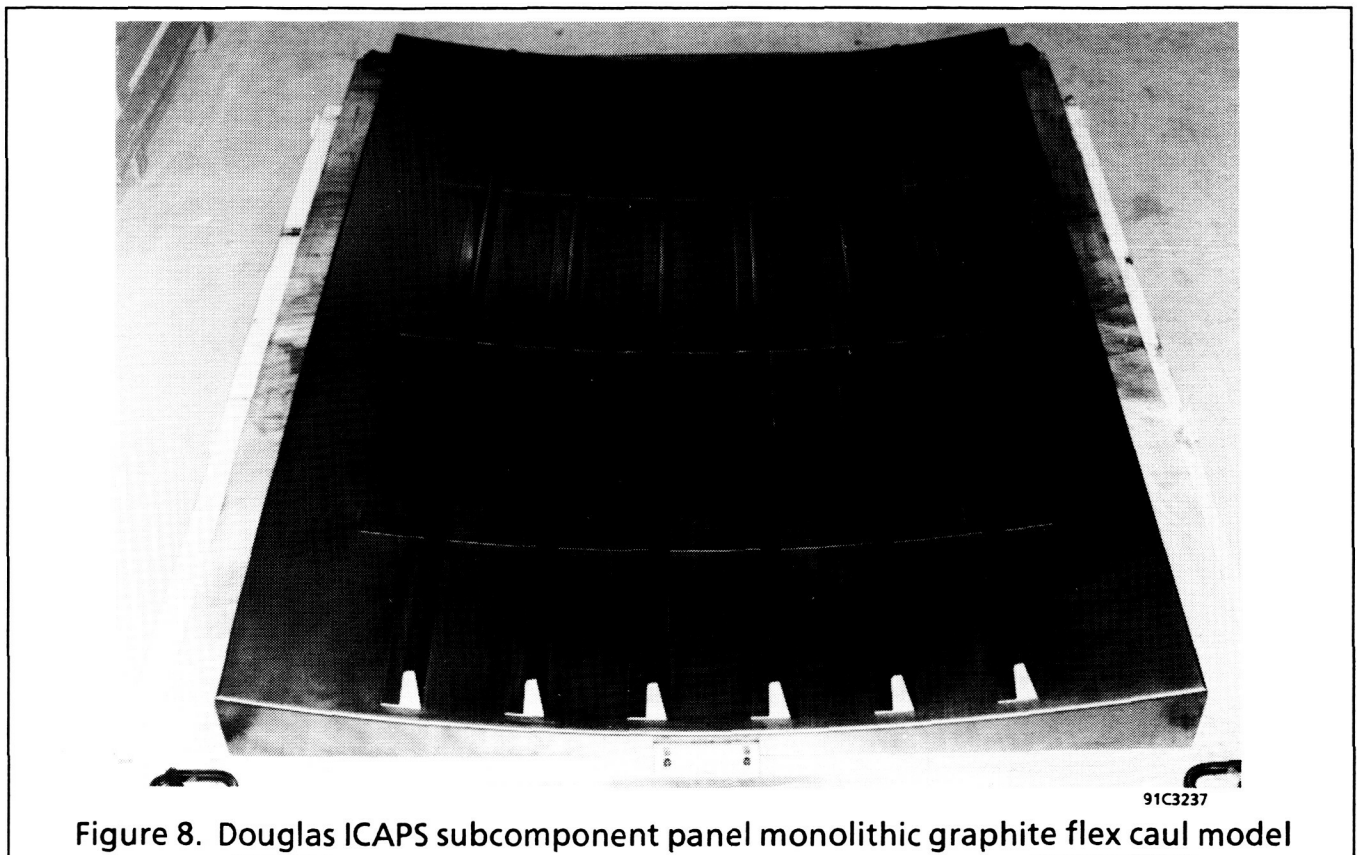


Figure 8. Douglas ICAPS subcomponent panel monolithic graphite flex caul model

The close tolerance stringer spacing achieved on both the flat element panels (Figure 9) and the 118-in. radius subcomponent panels can be attributed to use of the molded IML caul. The flex caul also produced excellent quality stringers that were near net dimension after cure tool removal and required very little deburring or trimming.

The fabrication process used for the tow-placed ICAPS panels was simple, easy to duplicate, and proved to be very low risk. The 12-ply skin panels were tow placed on an aluminum mandrel and transferred to the OML cure mold. The panel skin was then aligned to reference marks on the OML mold. The shear tee doublers were located to the skin IML, again by aligning to marks on the tool. The hand laid up J-stringers were fitted with a machined metal stringer cure mandrel (Figure 10) and the stringer/cure mandrel assembly

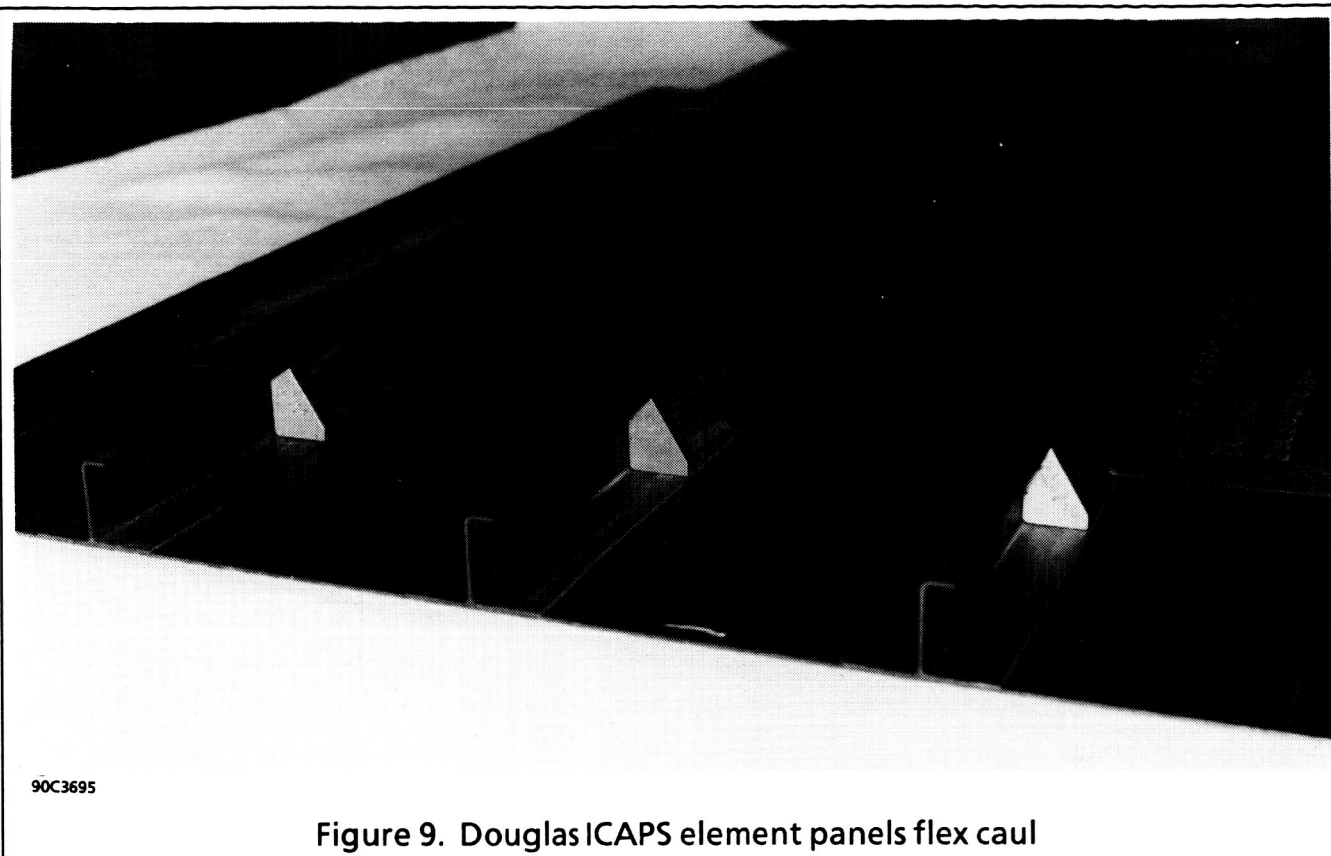


Figure 9. Douglas ICAPS element panels flex caul

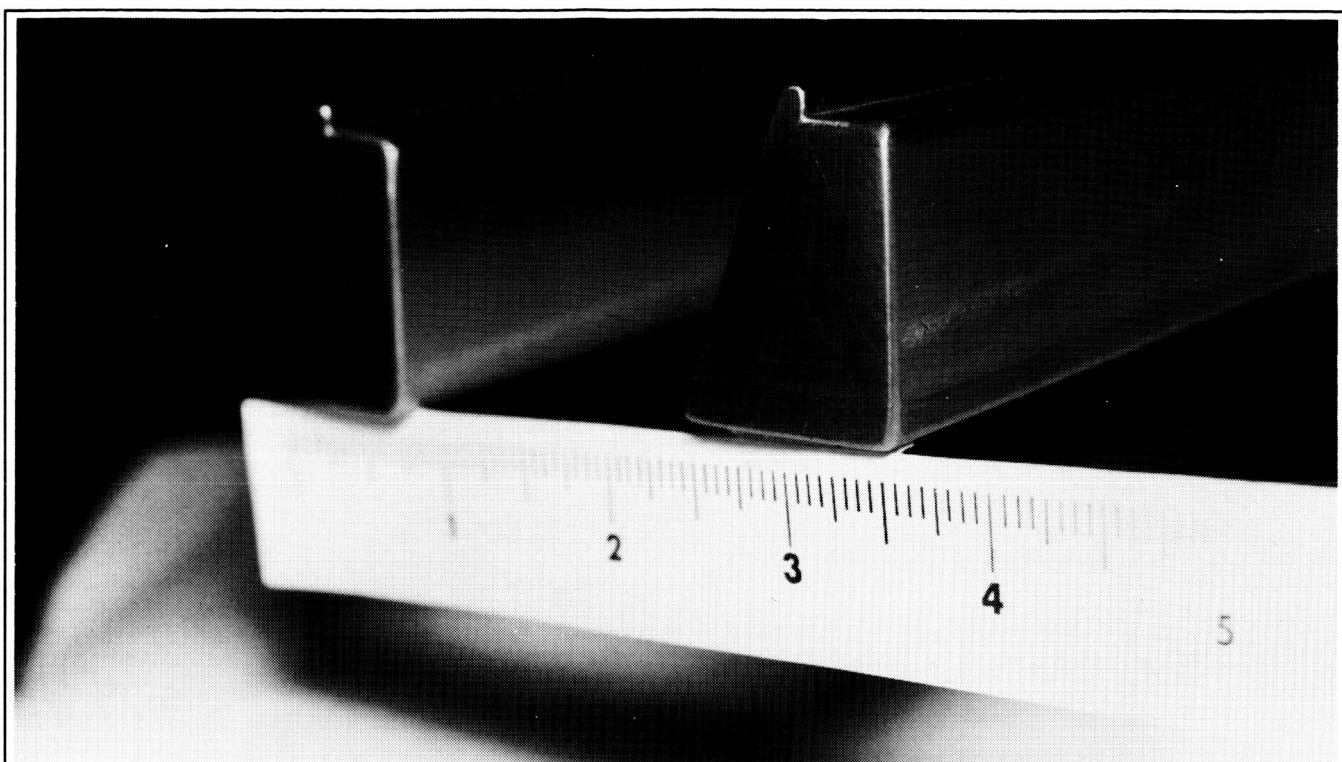
was positioned to the IML of the panel skin. Stringers were positioned using alignment marks on the tool. The molded graphite flex caul was then located over the skin/stringer assembly and pressed down to the skin IML. Pressing the flex caul down corrects any error in stringer position. The completed assembly was vacuum bagged and cured in the autoclave.

As the autoclave temperature increases and resin viscosity decreases, the autoclave pressure on the flex caul holds the stringers in proper position with a very close spacing dimensional tolerance.

After cure, the assembly was debagged and the tooling pieces were removed. The flex caul came off the panel with no problems. The stringer cure mandrels were removed by using T handles that screw into the sides of the cure mandrels. Removal of the stringer tools was not a problem. The panel was deburred and trimmed to net dimension (Figure 11).

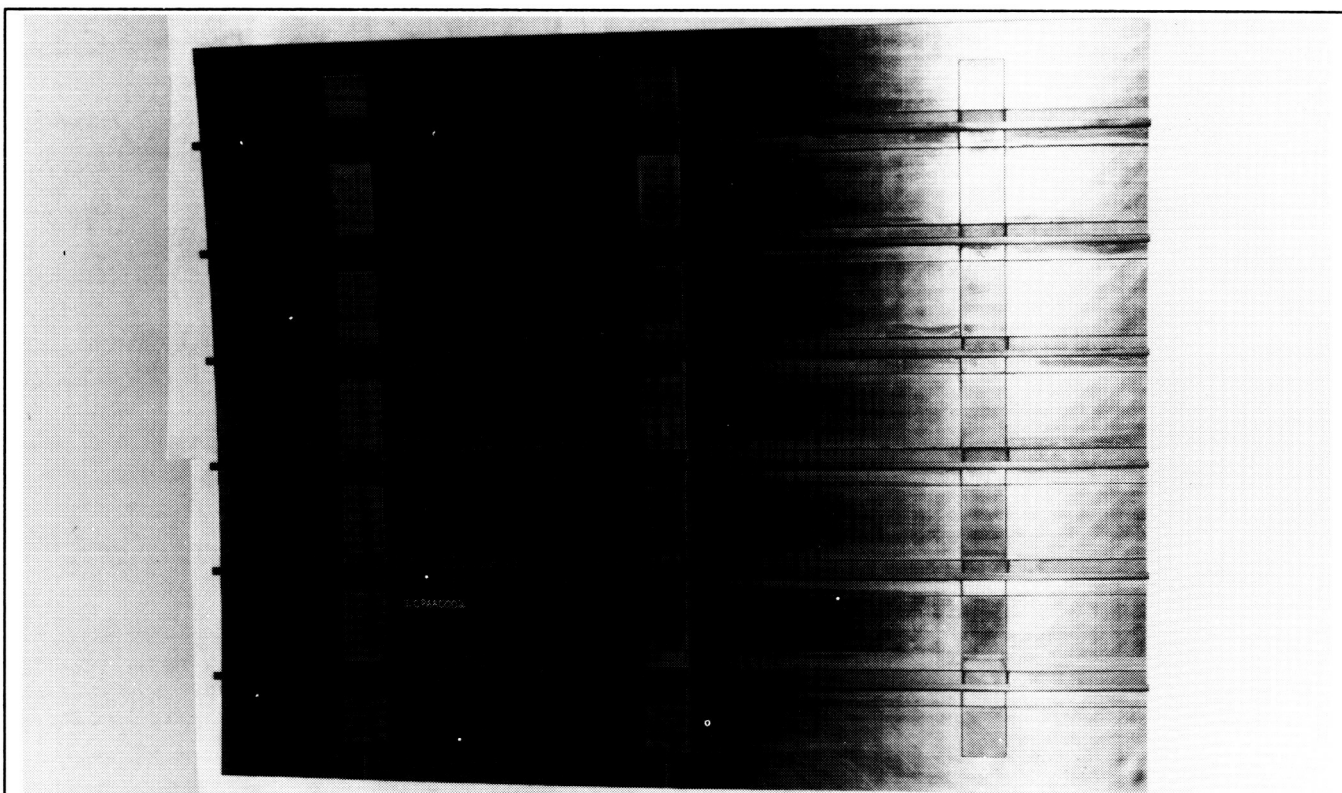
Dimensional and ultrasonic inspections were performed on each panel. No problem areas were discovered during NDI and overall quality of the panels was excellent.

The process used in fabrication of these panels was unique in its simplicity and successfully accomplished all objectives of this program. The process takes advantage of the low cost potential of the automated tow placement process and combines it with a low risk assembly and cure process. We believe our process can be easily adapted to larger fuselage panels and see very few problems in this scale up.



91C1544

Figure 10. Douglas ICAPS subcomponent panel stringer cure mandrel



91C4088

Figure 11. Douglas ICAPS subcomponent panel

## Service Tough Composite Structures Using the Z-Direction Reinforcement Process

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P. 7

## SUMMARY

Foster-Miller has developed a new process to provide through thickness reinforcement of composite structures. The process reinforces laminates locally or globally on-tool during standard autoclave processing cycles. Initial test results indicate that the method has the potential to significantly reduce delamination in carbon-epoxy. Laminates reinforced with the z-fiber process have demonstrated significant improvements in mode I fracture toughness and compression strength after impact. Unlike alternative methods, in-plane properties are not adversely affected.

## INTRODUCTION

Advanced composite materials such as carbon/epoxy are prone to delamination or splitting between the plies caused by handling or service impacts. Delamination is often internal and is difficult to detect even with sophisticated instrumentation. Critical components must be designed for conservative stress levels because of the possibility of an undetected delamination. Preventing delamination or limiting propagation under load would save weight through higher design allowables, and would reduce inspection, repair and replacement costs.

Several approaches have been evaluated for improved toughness of composite structures. Approaches evaluated to date often incur substantial cost to implement and/or result in significant reductions to in-plane properties.

Foster-Miller has developed a new approach to through thickness reinforcement called the z-fiber process. As will be shown in the sections which follow, this process has several desirable aspects including:

- No-dissimilar materials; through thickness reinforcements can match in-plane materials
- On-tool reinforcement with one side access
- Provides reinforcement locally or globally
- No specialized equipment required
- Compatible with existing tooling and process materials, equipment, and procedures
- Eliminates manufacturing steps (stitching or fastener installation)
- Potential for reduced fabrication, inspection, repair, and replacement costs
- Does not introduce any new or unqualified materials
- Increases interlaminar fracture toughness several times
- Significantly reduced delamination areas created by impact
- Minimal loss of in-plane strength

## THE FOSTER-MILLER Z-FIBER PROCESS

To meet the need for control of delamination, Foster-Miller has developed a new process for inserting through thickness fibers. The process converts a 2D prepreg layup to 3D on-tool with little or no change to standard cure cycles. The process is illustrated in figure 1. A foam prefoam containing small diameter rigid fibers is placed on top of the prepreg layup on-tool. During autoclave cure, the combination of heat and pressure compacts the foam which transfers the fibers into the composite. The through thickness

fibers are elastically supported by the foam to prevent buckling during insertion. After cure, the foam residue is removed along with the bleeder and release ply.

The foam preforms are produced by specialty machinery produced by Foster-Miller which can be programmed to insert fibers in any desired pattern and spacing. The foams can be produced in a variety of thicknesses, fiber densities, and fiber materials; and can be thermoformed to conform to curved surfaces. The process can be used to attach secondary structures during the co-cure as an alternative to metal fasteners (figure 2).

Foster-Miller has demonstrated the effectiveness of this technology with several different prepreg systems. Among these are AS4/3501-6, IM7-8551, and Fiberite K641 phenolic for carbon-carbon applications. The process can also accommodate several rigid rod materials as through thickness reinforcements. SiC, Boron, 30MSI carbon/epoxy, and P-100/epoxy (for thermal applications) rods have been evaluated as reinforcement materials to date. 30MSI carbon/epoxy has demonstrated the best fracture improvements in epoxy systems.

The ability of the foam to provide elastic stability to the rods allows the use of small diameter fibers. Our baseline reinforcement is 0.006 inches in diameter and is inserted at an areal density of 0.5% by area or 200 pins per square inch. The small diameter of the fibers and the ability to insert them at the point of minimum viscosity of the matrix system prevents significant fiber damage which can lead to in-plane property degradation. Small diameter fibers also maximize the laminate thickness to fiber diameter ( $l/d$ ) ratio which is important in thin laminates where the primary failure mode is fiber pull-out.

#### MECHANICAL PROPERTY EVALUATION

The mechanical property evaluation was conducted to determine what benefits in fracture toughness can be obtained with the z-fiber process and at what cost to in-plane property data. Static tension and compression and mode I fracture properties were evaluated.

Effects of z-fiber reinforcement on static tension properties are reported by Table I. Testing was independently conducted by Rohr Industries on 0/90 woven AS4/3501-5A fabric laminates. Baseline 0.006 inch diameter 30 MSI carbon/epoxy rod stock at 200 fibers per square inch was used. Reinforced panels exhibited 98% of the strength and modulus of control panels.

Effects of z-fiber reinforcement on compression properties is reported by Table II. Testing was conducted by LTV Aircraft Products using a 4 inch x 12 inch specimen in a standard CSAI test fixture. Two different diameter reinforcement materials were evaluated. In both cases reinforced laminates retained all of their strength compared to unreinforced laminates.

Improvements in mode one fracture toughness were determined using a double cantilever beam (DCB) test specimen. In this test, the load required to propagate a fracture in peel is plotted against the tensile testing cross head extension (figure 3) of the machines. The load is reduced to zero after the crack length travels 0.10 inches along the specimen length. Loading and unloading are then repeated for several 0.10 inch crack increments. The area under the curve for each individual area (represented by cross hatch in figure 3) is used to determine mode one fracture energy  $G_{IC}$ . The data plotted in figure 3 are for AS4/3501-6 laminates and compare unreinforced specimens with specimens reinforced with boron and carbon/epoxy rod stock. Boron and carbon/epoxy reinforced specimens exhibited over two and seven times, respectively, the fracture toughness of control laminates. The boron reinforced specimens failed by fracture of the boron fiber. This is believed to be a result of the brittle nature of boron in bending. Carbon/epoxy reinforced laminates failed from fiber pull out of the transverse fibers. This suggests if a better bond between the transverse fibers and the matrix can be obtained, or if a high percentage of z-fibers is used, even higher fracture values can be expected. A summary of the effect on mode one fracture properties for 3501 and 8551 laminates is shown by figure 4.

## IMPACT TESTING

Testing of AS4/3501-6 control versus reinforced laminates was conducted both with low velocity (drop weight) and medium velocity (simulated hail shot) impact.

Low velocity impact was evaluated using a 4 inch by 12 inch compression strength after impact (CSAI) specimen. The impact energy selected for testing was 20 ft•lbs (approximately 1000 in•lbs/inch). Two rod stock diameters, 0.006 and 0.008 inch diameter, were evaluated against controls. Testing was conducted by LTV Aircraft Products and the results are depicted by Table III. With both rod stocks close to 50%, improvement in CSAI was achieved.

Higher velocity impact was investigated using hail shot testing conducted by Boeing Military Airplanes. Eight inch square panels 0.125 inches in thickness were reinforced over a 5.25 inch square area about the center of the panels (figure 5). Two types of reinforcement patterns were investigated; reinforcement over the entire area, figure 5a, and reinforcement in a grid pattern, figure 5b. Areas of reinforcement contain 200 0.006 inch diameter fibers per square inch. The panels were mounted in a picture frame and impacted with 1.0 inch diameter hail balls at approximately 500 ft/sec. The resultant area of delamination for each panel condition was determined and is represented by figure 6. Area and grid reinforced panels, respectively, exhibited 44% and 57% less delamination area versus controls for the same impact level. It is theorized that grid panels were more effective because of the higher energy required to initiate delamination as the fracture approaches each group of fiber rows.

## CONCLUSIONS

The effectiveness of the z-fiber process in limiting delaminations and improving fracture toughness were clearly demonstrated through the testing conducted. Also demonstrated was the advantages of this process over alternative technologies in the areas of versatility, cost-effectiveness, and limit in in-plane property degradation. In summary the Foster-Miller z-fiber process demonstrated in AS4/3501-6 laminates:

- Greater than 7.5 times increase in mode one fracture toughness
- 50% increase in CSAI
- 45-55% decrease in delamination area when subjected to 80 ft•lb hail impact
- No decrease in in-plane compression strength properties
- 98% retention of in-plane tension strength and modulus

The z-fiber process increases service toughness without increasing weight and with no introduction of dissimilar or new materials. Manufacturers of composite structures will ultimately purchase preforms which can be used for through thickness reinforcement. There will be no requirement for special machinery, keeping application costs at a minimum.

## ADDITIONAL APPLICATIONS

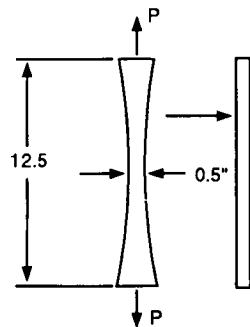
The z-fiber process is being evaluated for applications ranging beyond service toughness. Such applications include tailoring of through thickness thermal conductivity, attachment of co-cured structures, brazed attachment of carbon-carbon structures, sandwich core constructions, and improvements in interlaminar tension strength.

## ACKNOWLEDGMENTS

Work conducted for this paper was performed under a Small Business Innovative Research (SBIR) contract #N60530-88-C-0370 from the Naval Weapons Center China Lake. The technical monitors have been Mr. Burt Stull succeeded by Mr. Richard Bott. In addition, cost sharing in the terms of test services and technical support have been provided by LTV Aircraft Products, Boeing Military Airplanes, and Rohr Industries. Special thanks to Glen Williams and Scott Norwood of LTV, Jamie Childress of BMA, and Chris Saddler of Rohr for their contributions.



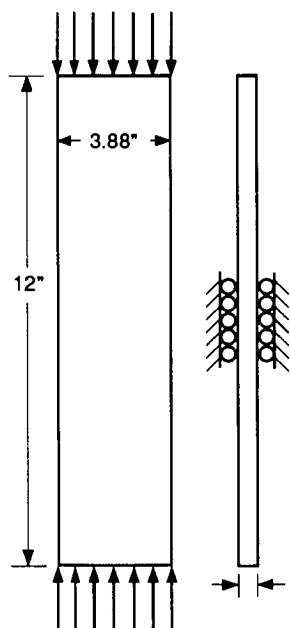
TABLE I. - STATIC TENSION DATA



|                | CONTROLS<br>UNREINFORCED | REINFORCED | % OF CONTROL<br>RETAINED |
|----------------|--------------------------|------------|--------------------------|
| STRENGTH (ksi) | 127                      | 124        | 98                       |
| MODULUS (Msi)  | 11.9                     | 11.7       | 98                       |

61-FA-1

TABLE II. - COMPRESSION DATA



| % AREA<br>CARBON/EPOXY<br>Z-FIBER<br>(pins/in. <sup>2</sup> ) | Z-FIBER<br>DIAMETER<br>(in.) | FAILURE<br>STRESS<br>(ksi) | % OF<br>CONTROL<br>RETAINED |
|---------------------------------------------------------------|------------------------------|----------------------------|-----------------------------|
| NONE                                                          | NONE                         | 92,080                     | ---                         |
| 0.5 (196)                                                     | 0.006                        | 92,936                     | 101                         |
| 0.5 (100)                                                     | 0.008                        | 93,968                     | 102                         |

MATERIAL: AS4/3501-6  
LAMINATE ORIENTATION:  
[±45/02/90±45/02/90±45/02/90±45/0/±45]S  
OR 37.5/50/12.5 PERCENT 0/45/90

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TABLE III. - CSAI DATA

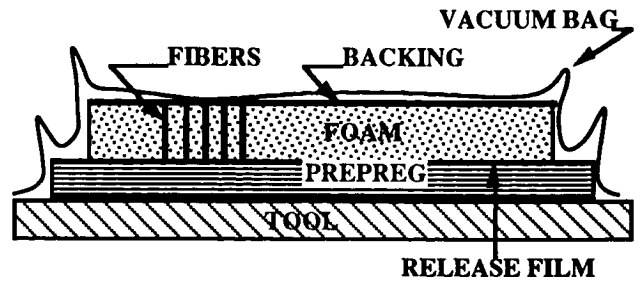
| %<br>Z-FIBER<br>(pins/in. <sup>2</sup> ) | Z-FIBER<br>DIAM<br>(in) | FAILURE<br>STRESS<br>(ksi) | %<br>INCREASE |
|------------------------------------------|-------------------------|----------------------------|---------------|
| NONE                                     | NONE                    | 29,267                     | —             |
| 0.5                                      | 0.006                   | 43,738                     | 49.4          |
| 0.5                                      | 0.008                   | 43,000                     | 46.9          |

MATERIAL: AS4/3501-6  
LAMINATE ORIENTATION:  
[±45/02/90±45/02/90±45/02/±45/0/±45]S  
OR 37.5/50/12.5 PERCENT 0/45/90

61-FA-4

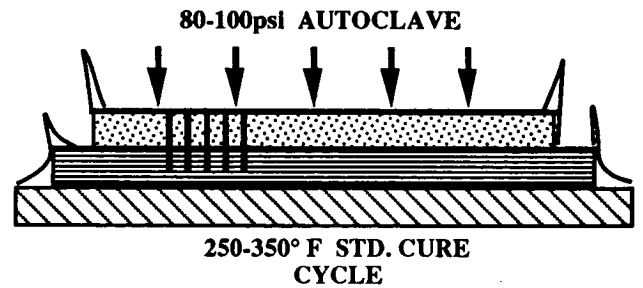
1. PLACE RELEASE FILM AND Z-PREFORM ON TOP OF PREPREG LAYUP AND THEN BAG

(1)



2. DURING STANDARD CURE CYCLE, HEAT AND PRESSURE COMPACT FOAM - FORCING THE FIBERS THROUGH THE LAMINATE

(2)



3. REMOVE COMPACTED FOAM AND DISCARD ALONG WITH BAGGING MATERIALS

(3)

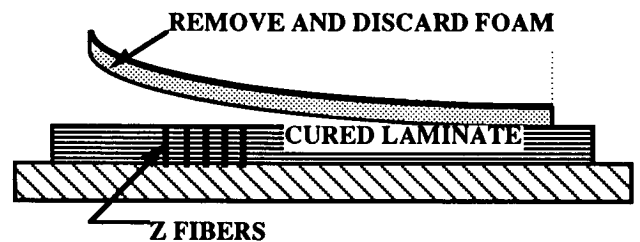


Figure 1. - Z-Fiber Process

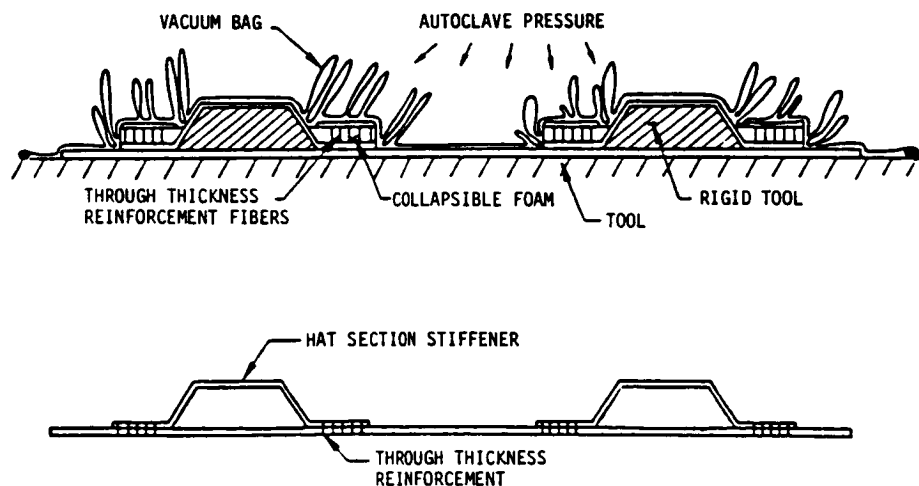


Figure 2. - Stiffener Attachment During Co-Cure



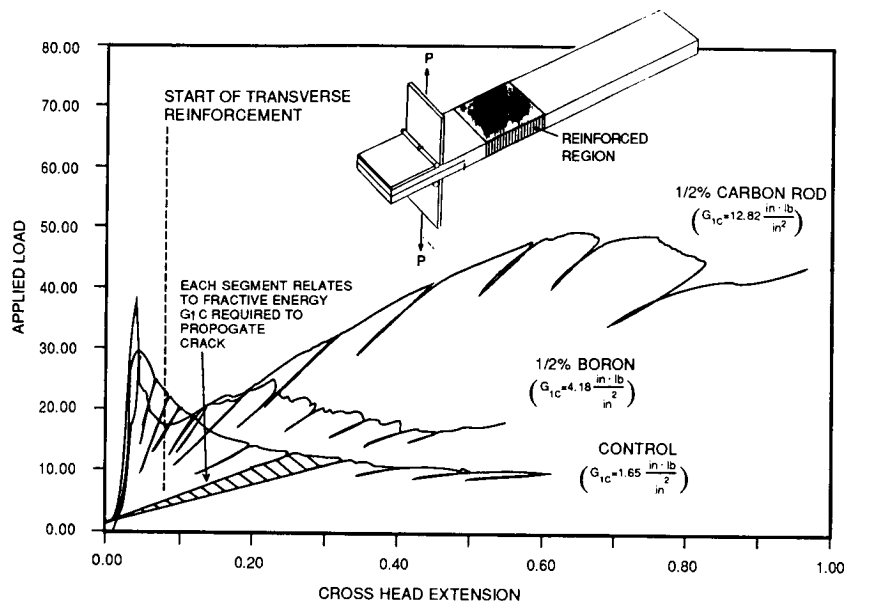


Figure 3. - DCB Plots of Various Reinforcement Materials

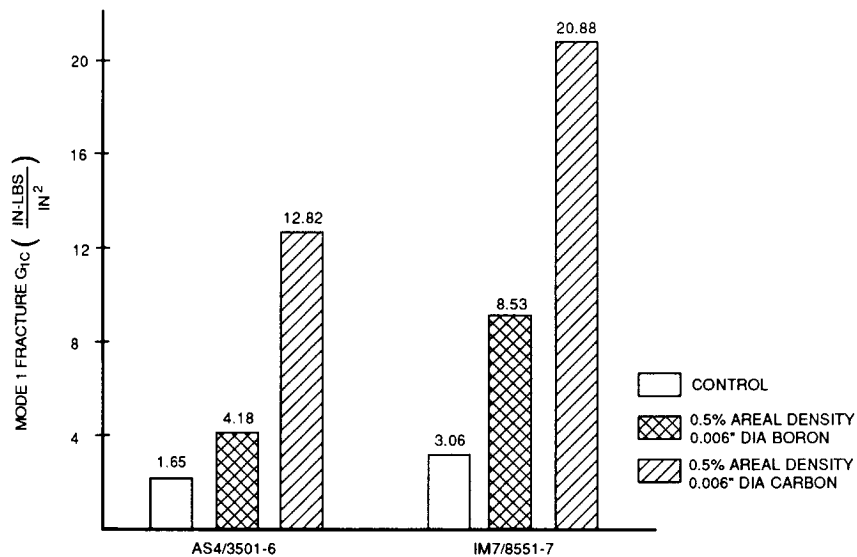


Figure 4. - Summary of Mode One Fracture Testing

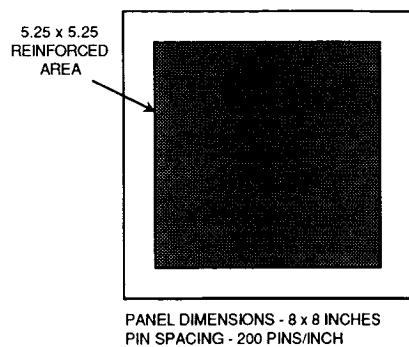


Figure 5a. - Area Reinforced  
Hail Shot Panel

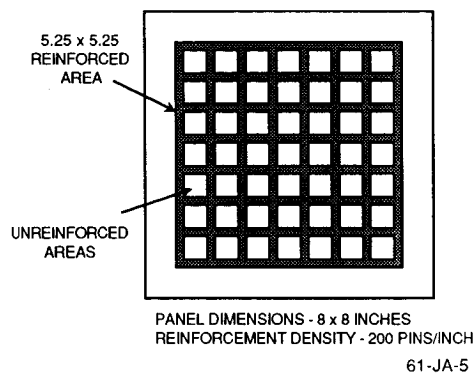
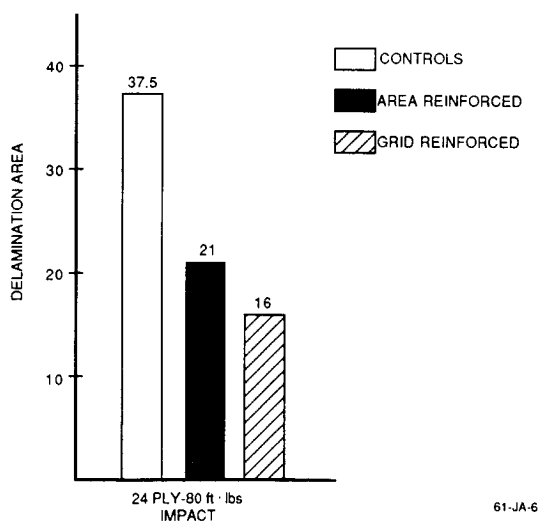


Figure 5b. - Selectively Reinforced  
Hail Shot Panel

Figure 5. - Hail Shot Panels

61-JA-5



61-JA-6

- QAU SI - ISOTROPIC
- 0.5% BY AREA - 0.006" DIAMETER CARBON ROD STOCK
- 1.0" DIAMETER HAIL 500 ft/sec
- DATA REPRESENTS SMALL SAMPLING; 2 DATA POINTS MINIMUM, 4 DATA POINTS MAXIMUM

Figure 6. - Trends in Hail Shot Impact Delamination Area

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THROUGH-THE-THICKNESS<sup>R</sup> BRAIDED COMPOSITES<sup>\*</sup>  
FOR AIRCRAFT APPLICATIONS

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Material and structural specimens of Through-the-Thickness<sup>R</sup> braided textile composites have been tested in a variety of experiments. The results have demonstrated that the preform architecture provides significant payoffs in damage tolerance, delamination resistance, and attachment strength. This paper describes the braiding process, surveys the experimental data base, and illustrates the application of three dimensional braiding in aircraft structures.

### INTRODUCTION

Through-the-Thickness<sup>R</sup> braided textiles are a new class of composite material which significantly improve material toughness. The technique produces seamless, thick textiles by continuous intertwining of fibers. Materials made from this type of three-dimensional (3-D) reinforcement provide advantageous structural performance especially suitable for aircraft applications.

The use of composite materials in aircraft structures has become widespread in recent years because of their high strength and stiffness relative to weight. Laminated, two-dimensional (2-D) composites are the current state-of-the-art but can be weakened by delamination caused, for example, by holes, cut-outs, or foreign object damage.

For the same weight of material, Through-the-Thickness<sup>R</sup> braided composites have the following advantages compared to 2-D laminates:

- 20% increase in shear strength,
- 50% increase in shear stiffness,
- 40% improvement in residual compressive strength after impact,
- 50% higher tensile strength near cut-outs, and
- 300% greater rib/skin attachment strength.

Components with features such as stiffeners or cut-outs, thick sections, or structures carrying shear loads, or those which have exposure to debris are applications where these advantages can translate into weight savings.

\*Through-the-Thickness<sup>R</sup> is a registered trademark of the Atlantic Research Corporation.

## THROUGH-THE-THICKNESS<sup>R</sup> BRAIDING

Composite materials with reinforcements oriented in two directions are the state-of-the-art for highly loaded structures such as those found in aircraft applications. These materials are fabricated by lamination of woven fabric plies, lay-up of bias or unidirectional tape, or by filament winding. This construction allows for the individual layers to be oriented so that the stiffness and strength of the reinforcement can be aligned in the direction of the applied loads. While tailored for expected loading conditions, the layered, laminated nature of the construction relies upon a relatively weak bonding agent to transfer the stresses from layer to layer and around cut-outs or where damage has occurred. This construction therefore, is susceptible to delamination and crack propagation.

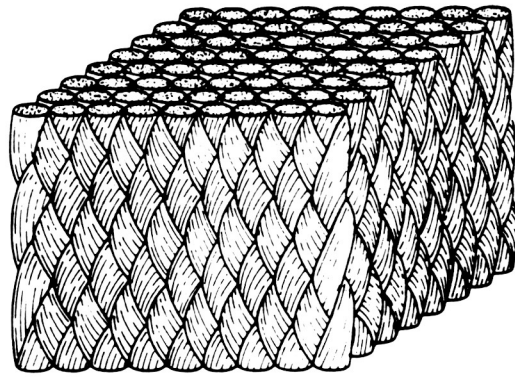
3-D, Through-the-Thickness<sup>R</sup> braiding was developed by Atlantic Research Corporation to eliminate the possibility of delamination. This is accomplished by continuously intertwining the reinforcing fibers into the seamless, nonlaminated structure<sup>1</sup> shown in Figure 1. The 3-D architecture eliminates planes of delamination with a moderate sacrifice in the in-plane properties. Additionally, the presence of crack-arresting fibers in every orientation markedly reduces the propagation of damage through the structure.

While other 3-D textiles such as multi-ply woven, layer interlock and stitched fabrics also have been developed to prevent interply failure, Through-the-Thickness<sup>R</sup> braiding is the only textile technique which, by its unique architecture, completely eliminates the delamination prone layered structure. In addition, 3-D braiding eliminates bonding or the use of mechanical fasteners with its unique capability to fabricate complex shapes which totally integrate the reinforcements, for example, between webs and flanges.

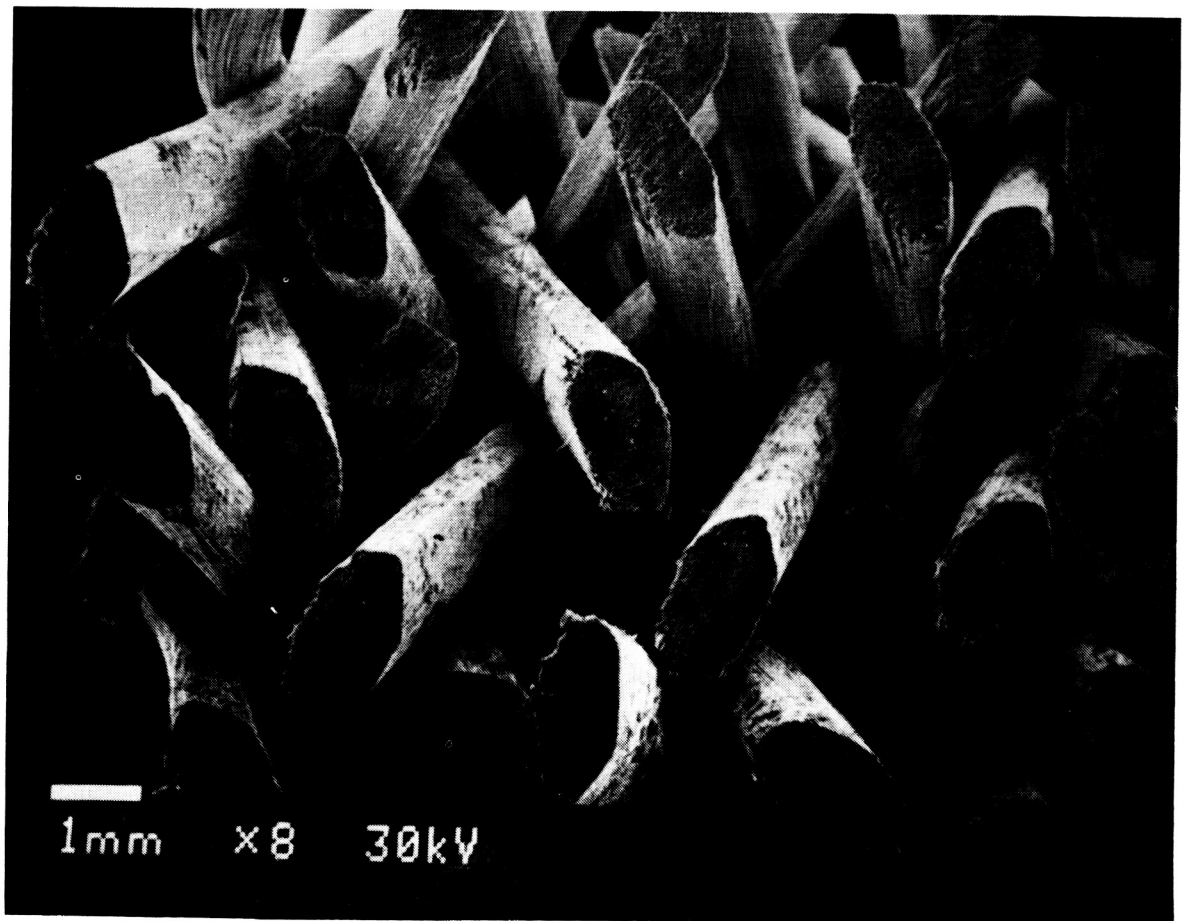
Figure 2 illustrates the operation of an automated 3-D braiding machine. The eight foot diameter machine's scale is apparent from the photograph in Figure 3. The machine has a capacity of 3,168 bobbins, each bobbin holding approximately 30 meters of 12K tow fiber. This braider can produce textile sheets 60 inches in width by 0.25 inches in thickness. It is controlled by an Omron programmable logic controller (PLC) networked with two other 3-D braiders into an MS-DOS based computer. The braiding control software allows input of braid plans, schedules and assigns jobs to individual machines, maintains a quality data history for each job, and diagnoses machine faults.

## IMPACT PERFORMANCE

Unintentional impact by foreign objects can weaken composite structures. Runaway debris is a frequent source of damage to the underside of any aircraft and battle damage occurs to military aircraft. 3-D braided composites are extremely tolerant of impact damage.



(a) Intertwined Fibers with No Planes of Lamination.



(b) SEM Photograph of Braided Microstructure.

Figure 1. Through-the-Thickness<sup>R</sup> Braid Structure.

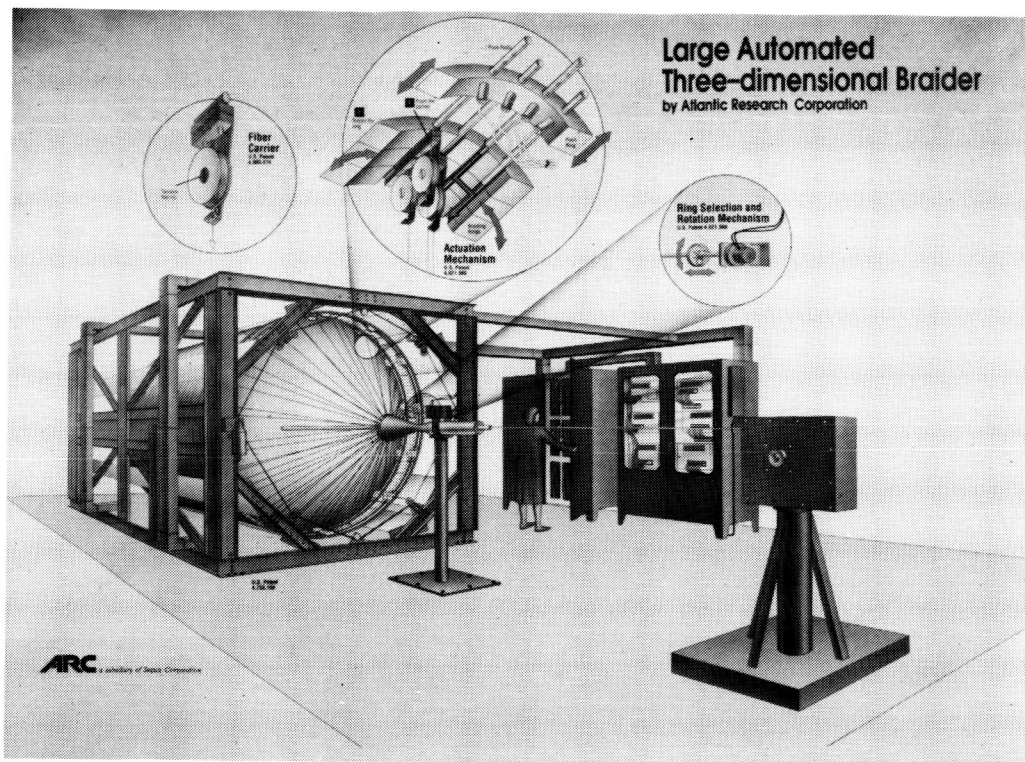


Figure 2

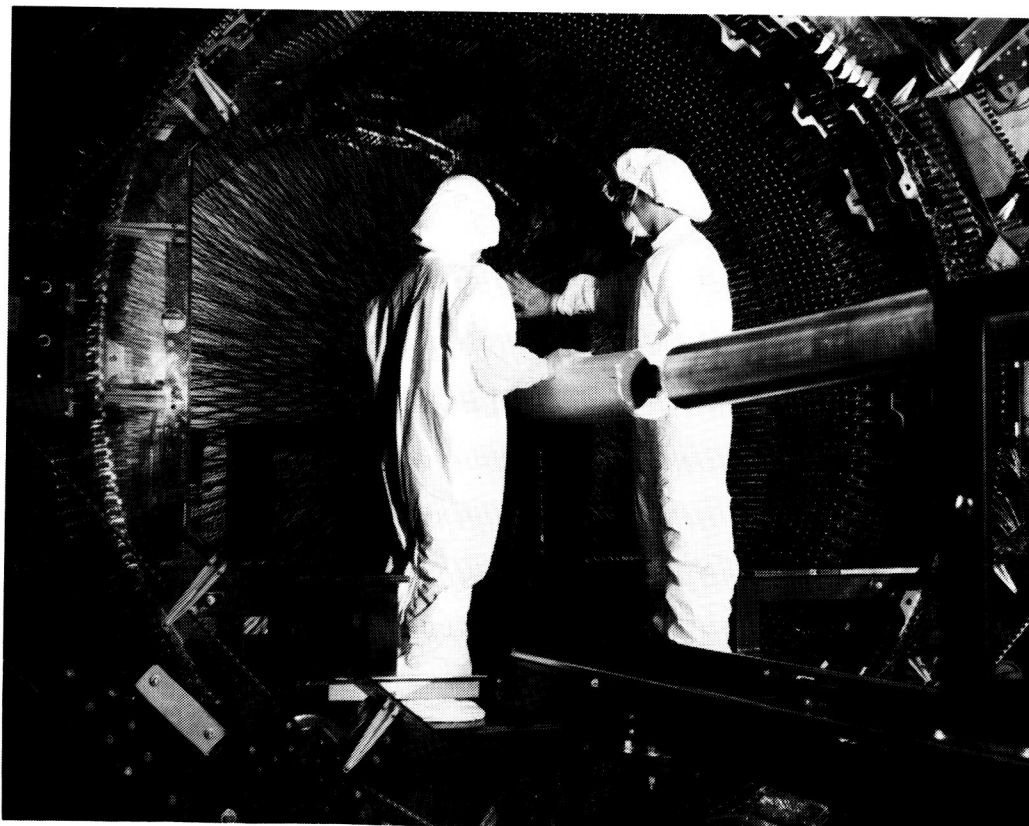


Figure 3

The Naval Air Development Center<sup>2</sup> (NADC) assessed the damage tolerance of 3-D braided composite skins by performing instrumented impact tests of two styles of 3-D braid and a comparison 2-D laminate. The braided skins were fabricated from Celion 12000 carbon fiber with the fibers oriented  $(\pm 20)_s$  for style #1 and  $(\pm 20/0)_s$  for style #2. [Note: The results of the NADC studies are cited several times in this report. Reference to braid styles #1 and #2 is made to simplify the text and is the notation used by the NADC. This does not imply that ARC manufactures only these two styles of material.]

The braids were impregnated with an aerospace resin, Hercules 3501-6, and autoclave cured. The comparison laminate was manufactured from aerospace grade AS-1/3501-6 prepreg and autoclave cured. The stacking sequence for the 24 ply laminate was  $(\pm 45/0_2/\pm 45/0_2/\pm 45/0/90)_s$ . The average fiber volume for the three types of material varied between 50% and 55%. Plate specimens 4 inches by 8 inches were tested in an Effects Technology, Inc. ETI-8200 drop tower. Half of each plate was clamped about the edges, leaving a 3-inch square area for impacting and allowing two impact tests per plate.

Four replicate impact tests for each style braid and the comparison laminate were performed at three different impact energy levels: 4 ft-lb (approximate incipient damage), 10 ft-lb (approximate peak load) and 115 ft-lb (through penetration load). Impact energy is plotted versus damaged area in Figure 4. The figure also includes 32 ply AS-6 data from another source<sup>3</sup>. At impact levels above the incipient damage level, the 3-D braids are superior to the laminate in limiting the extent of damage.

The extent of damage directly reduces the strength of 2-D laminates. To verify the expected improvement in post-impact residual strength of 3-D braided composites, several compression after impact experiments were performed at ARC comparing 2-D woven and braided specimens with 3-D braided material. Table 1 describes the specimen construction; both  $\pm \theta$  and  $0 \pm \theta$  architectures were evaluated. Test results are shown in Figure 5. Specimen dimensions were 3 inches square by 0.2 inches thick. Impacts of 150- and 300-inch pounds were delivered by a calibrated, rail guided, free drop penetrator. The impact specimen was held by a 0.5 inch wide clamped edge. At 150-inch pounds surface damage was negligible. All tests were repeated four times.

Because of differences in fiber volume between the test specimens, the results presented in Figure 5 are normalized. Ultimate compressive strengths of undamaged specimens are tabulated in Table 1. Most of the specimens indicated equal drops in compressive strength at 150 inch pounds. However, at 300 inch pounds the 3-D braided specimens still retained that same amount of strength while the 2-D specimens suffered a dramatic reduction in load carrying capability.



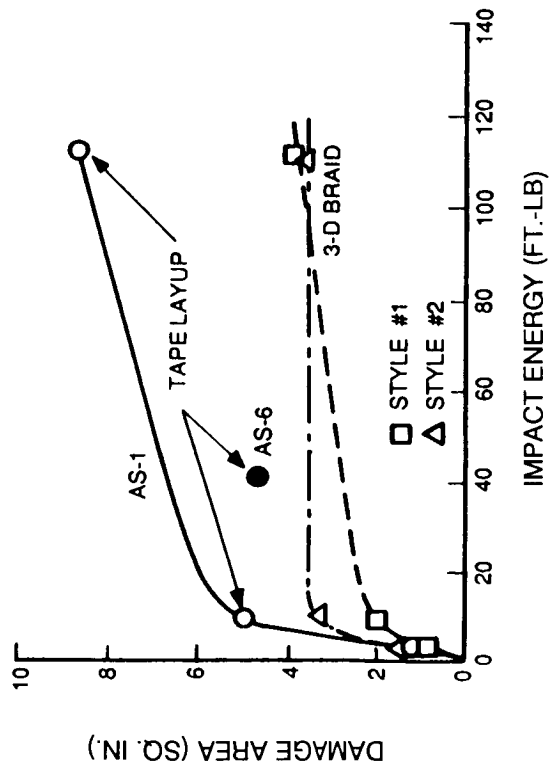


Figure 4. Damage Area Versus Impact Energy.

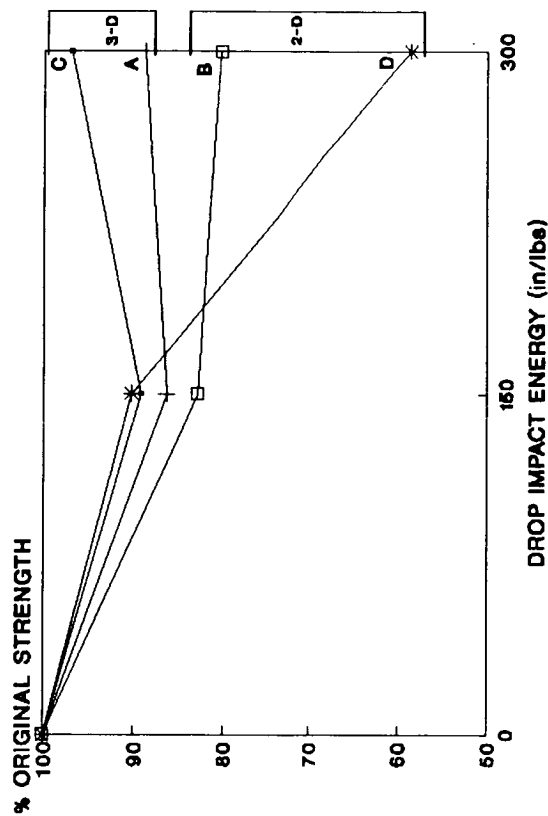


Figure 5. Compressive Strength Versus Impact Energy.

Table 1

# MATERIAL DATA FOR COMPRESSION AFTER IMPACT TESTING

| Material | Construction          | Fiber                                           | Process                          | Fiber Volume | Ultimate (undamaged) Compressive Strength |
|----------|-----------------------|-------------------------------------------------|----------------------------------|--------------|-------------------------------------------|
| A        | 3-D Braid             | AS 4 @ $\pm 45^\circ$                           | Press mold with Shell 9405 resin | 58%          | 21.7 ksi                                  |
| B        | 2-D Weave             | AS 4 @ $\pm 45^\circ$                           | Press mold with Shell 9405 resin | 66%          | 27.5 ksi                                  |
| C        | 3-D Braid             | G50-300 @ $0^\circ$<br>G40-600 @ $\pm 45^\circ$ | RTM with Shell 9405 resin        | 44%          | 26.1 ksi                                  |
| D        | 2-D Braid/<br>UD Tape | G50-300 UD<br>G40-600 @ $\pm 45^\circ$          | RTM with Shell 9405 resin        | 46%          | 59.2 ksi                                  |

## POST-IMPACT FATIGUE

The damage tolerant nature of 3-D braided composites is unaffected by repeated flexure. In studies performed by the NADC, David Taylor Naval Ship Research and Development Center (DTNSR&DC), and Virginia Polytechnic Institute and State University (VPI), the ultimate strength and stiffness of fatigued components were shown to be essentially unchanged from initial values.

Fatigue properties of 3-D braided columns were investigated by NADC<sup>4</sup>. Figure 6 plots percentage of initial buckling and failure strengths for braided Celion 12000/Hercules 3501-6 channel and cruciform sections after one and two million load cycles at 70% of the crippling strength. The figure shows that the ultimate failure strength is virtually unchanged after two million cycles. There is some drop-off of buckling strength but the value remains constant over the fatigue life.

Several experiments have been performed to evaluate the behavior of 3-D braided composites under repeated loading after sustaining impact damage. Performance usually is evaluated in terms of an increase in measured deflection or as a loss of stiffness.

DTNSR&DC<sup>5</sup> conducted fatigue tests on a 3-D braided marine propeller blade. A constant centrifugal load of 2,160 lbs was applied at the center of gravity and a simulated hydrostatic load of 1,650 lbs was applied perpendicular to the center of pressure. The perpendicular load was cycled at a frequency of 5 hertz at an amplitude of 50% of the mean hydrostatic load. At 3.25 million cycles, the blade was impacted at the maximum bending stress location (near the root) with 400 ft-lbs of energy. Local damage was sustained in the form of a crack through the thickness of the blade. On reapplication of cyclic loading, damage did not grow in area during an additional 2.5 million cycles of post impact fatigue. As shown in Figure 7, the strain response remained linear and tip deflections were unchanged.

VPI tested 3-D braids in static and post impact fatigue<sup>6</sup>. The test conditions and results, plotted as normalized stiffness versus normalized life, are given in Figure 8. In order to provide a comparison to an undamaged laminated material, the data from reference 7 was added to the graph. The results show that during the initial 70% of life, the fatigue response of the damaged and undamaged braids are nominally identical, with a slow drop-off in stiffness over time. In contrast, the undamaged laminate exhibits an initial and rapid drop in stiffness followed by a continuous but gradual degradation. Related impact tests also have been reported<sup>8,9,10,11</sup> by others with results consistent with those presented in this paper.

## OPEN HOLE TENSION/COMPRESSION TESTING

3-D braided materials are insensitive to the presence of holes in tension and compression stress fields. Aircraft bulkheads typically have holes and other cut-outs to

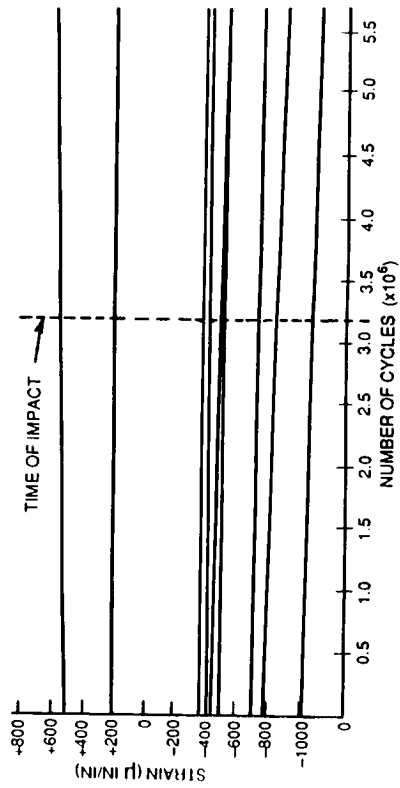


Figure 7. Strain Response of Marine Propeller Fatigued and Damaged.

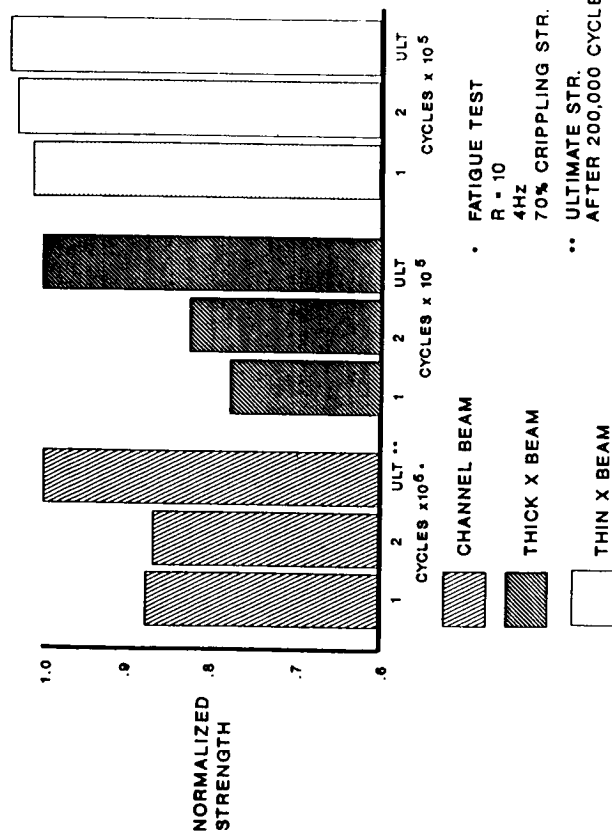


Figure 6. 3-D Braided Column Fatigue.

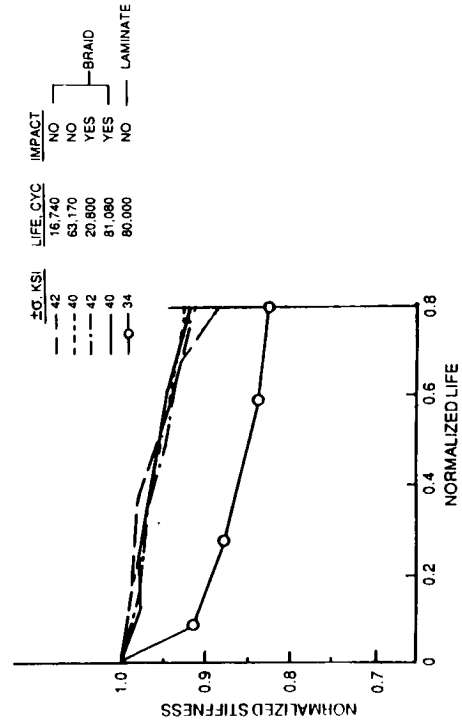


Figure 8. Stiffness Response to Fatigued and Damaged Specimens.

either provide openings for doors, windows, cables and tubing or to reduce weight. The presence of a hole or cut-out results in a local stress riser which magnifies the applied load.

A typical manufacturing approach for laminated composites is to form the hole rather than cut it, because a formed hole reduces the magnification factor<sup>12</sup>. However, such a formed hole increases both design and fabrication costs. In contrast, a hole can be cut or drilled in a 3-D braided composite without additional expense and without magnification of the applied tensile load.

The open hole test is a standard means of determining the material sensitivity to cut-outs. The NADC conducted these tests<sup>2</sup> in tension on the style #1 and #2 braided composites and the laminated composite described previously. The standard tensile specimen was a bar 9 inches long, 1 inch wide and 1/8 inch thick. A 1/4 inch diameter hole was drilled in the center of the bar, reducing the cross-sectional area by one-fourth. Five replicate tests were performed for each material and compared to five tests each of a solid (no hole) coupon.

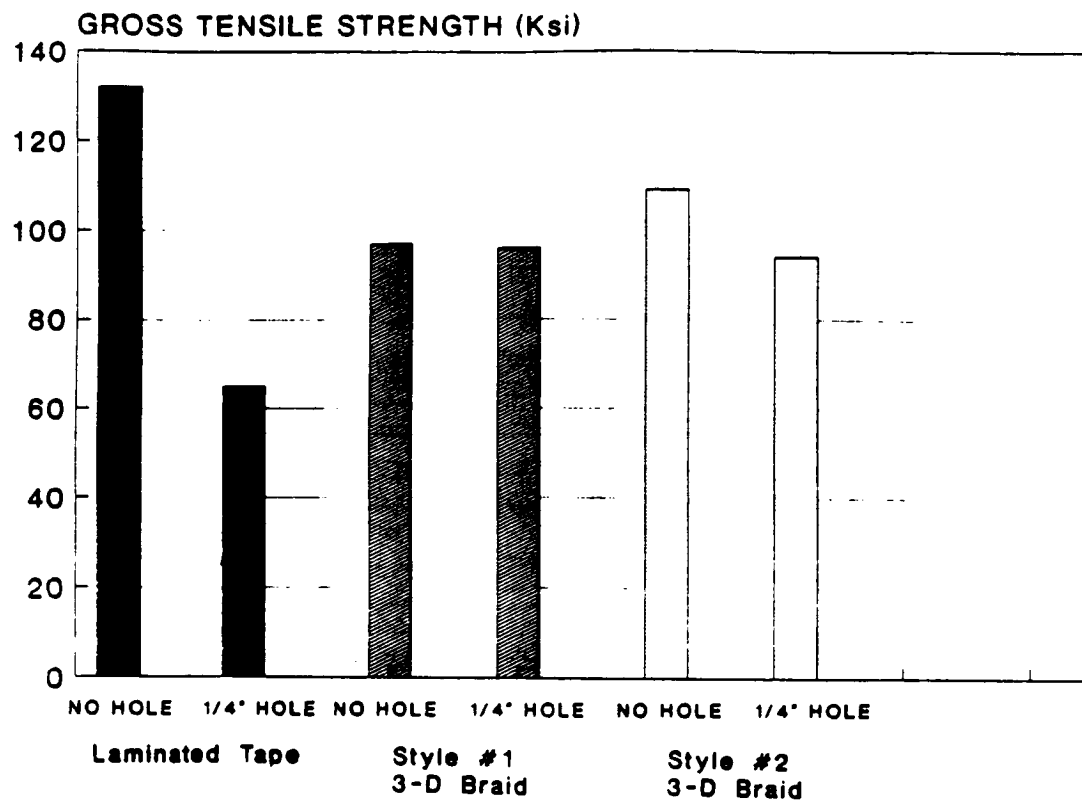
The average gross tensile stresses (load/no hole area) for each material are plotted in Figure 9. The results show that while the 2-D laminate is initially stronger, it incurs a 50% reduction in tensile strength in the presence of a hole. In contrast, the tensile strength of the style #1 braided material is reduced by less than 1% and by only 13% for style #2. This demonstrates that the 3-D architecture is relatively insensitive to holes and cut-outs of up to one fourth of the load carrying area in a tensile field.

ARC has conducted open hole compression testing on 2-D and 3-D braided architectures described in Table 2. Results are given in Figure 10. Five replicate tests were performed for each material, and test results are normalized to account for fiber volume and orientation differences. Compressive strengths of no hole coupons are reported in Table 2. The results are consistent with those reported by the NADC for tension. The 3-D braided architecture was clearly less sensitive to the presence of a cut hole than the 2-D braid.

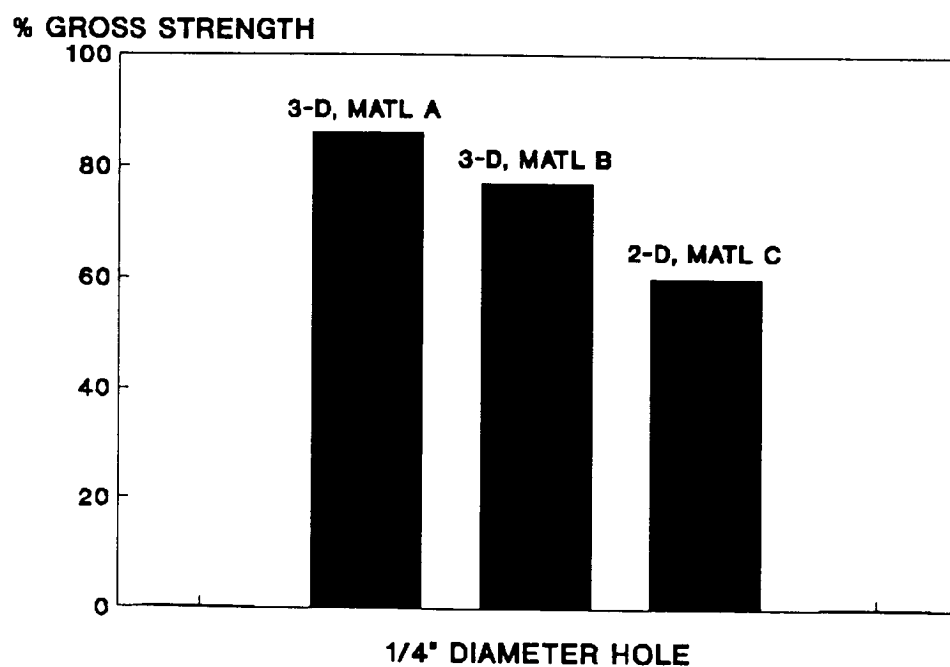
## STIFFENER PULL-OFF

Stiffness critical members depend on the cross-sectional inertia of built-up sections rather than the modulus of the sections alone. In this case, stiffness depends on the strength of the section connections and on the buckling resistance of the integrated component. 3-D braided skin panels with integral rib or hat stiffeners have significantly stronger skin to stiffener joints than bonded constructions.

A T-beam stiffener pull-off test was conducted by the NADC<sup>13</sup> to determine the failure modes and strengths of integrally braided rib to skin structures. The beam was braided using the style #2 pattern. Figure 11a illustrates the interlocking of the skin



**Figure 9. Open Hole Tensile Strength.**

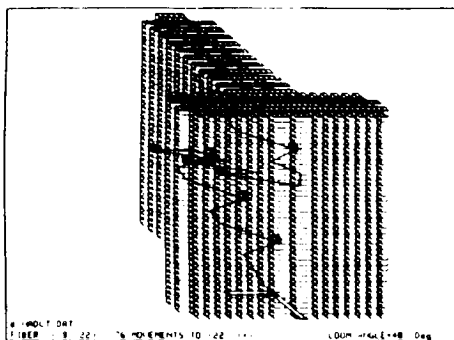
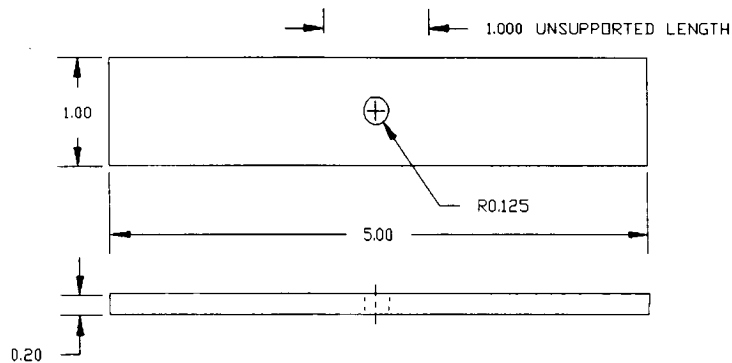


**Figure 10. Open Hole Compressive Strength.**

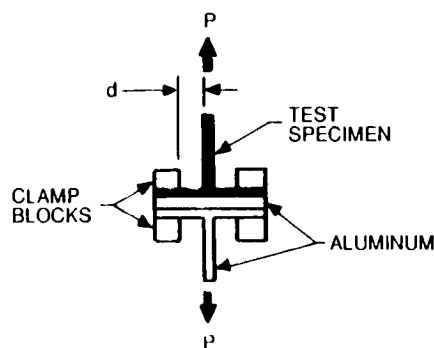
Table 2

## MATERIAL DATA FOR OPEN HOLE COMPRESSION TESTING

| <u>Material</u> | <u>Construction</u>    | <u>Fiber</u>                    | <u>Process</u>                   | <u>Fiber Volume</u> | <u>Ultimate (undrilled) Compressive Strength</u> |
|-----------------|------------------------|---------------------------------|----------------------------------|---------------------|--------------------------------------------------|
| A               | 3-D Braid              | 650-300 @ 0°<br>640-600 @ ± 45° | RTM with Shell 9405 resin        | 44%                 | 20.2 ksi                                         |
| B               | 3-D Braid              | AS 4 @ 0°<br>AS 4 @ ± 60°       | Press mold with Shell 9405 resin | 56%                 | 50.8 ksi                                         |
| C               | 2-D Braid/<br>U D Tape | 650-300 @ 0°<br>640-600 @ ± 45° | RTM with Shell 9405 resin        | 46%                 | 52.3 ksi                                         |



(a) Braided Integration of Rib &amp; Skin



(b) Rib Pull-Off Test Set Up

## RESULTS

| LOAD (lbs) | LINEAR LOAD (lbs/in) | FAILURE MODE |
|------------|----------------------|--------------|
| 373        | 447                  | STIFFENER    |
| 325        | 410                  | MIXED        |
| 190        | 170                  | SKIN         |

AVERAGE LINEAR LOAD FOR BONDED STIFFENER = 50 lbs/in

Figure 11. Stiffener Pull-Off Test.

and rib sections by tracing the path of a single fiber as it travels between them. All of the fibers are traveling in similar paths, resulting in a completely integrated structure.

The test set-up and results are illustrated in Figure 11b. Five tests of the T-beam were conducted with two of the tests using the same clamp spacing on the web. The rib was pulled in tension at a constant head speed of 0.05 inches per second. Because of higher rib attachment strengths, the initial tests resulted in transverse failure of the rib rather than the attachment. The clamp spacing was subsequently varied in an attempt to shift the failure location to the rib/skin intersection. No actual separation was achieved. Instead the failure site and mode were moved to a skin bending failure. Depending on the failure mode chosen, rib tension or skin bending, the load carried per running inch (distance between clamps) varied from 170 lbs/inch to 447 lbs/inch.

Lockheed<sup>14</sup> measured pull-off loads for a similar thickness but bonded T-beam of only 50.0 pounds per linear inch, with failure always occurring at the rib to skin bond. Comparison with the minimum linear load carrying capability of 170 lbs/inch obtained by NADC yields over a 300% improvement in stiffener attachment strength.

## MECHANICAL PROPERTIES

Material tests have been performed by Atlantic Research to characterize the strength and stiffness of 3-D braided composites using a variety of graphite fibers and epoxy resins. Some panels were braided to a  $(\pm 45)_s$  fiber orientation. Test specimens were cut to yield  $(0/90)$  on-axis and  $(\pm 45)_s$  off-axis properties. Other panels were braided with a quasi-isotropic orientation of  $(0\pm 60)_s$ . Five replicate tests were performed for each material in each loading condition. Tests were performed in tension and compression, on and off-axis, and in shear using the IOSIPESCU test configuration. The average properties for each material are reported in Table 3.

## COMPONENT DEMONSTRATIONS

Four structural components have been fabricated in sizes and shapes which go beyond the simple requirements of material testing coupons. These structures, Figures 12-15 illustrate stiffener to skin integration and forming of complex geometries. The structures also demonstrate a variety of processing methods.

The hat stiffened panel, Figure 12, was made from an ARC supplied preform by the NADC using a hot melt/autoclaving process with Hercules 3501-6 resin. The spar, Figure 13, was drape molded and vacuum bag cured at ARC using Shell 9405 resin. The J-stiffened panel, Figure 14, was fabricated in a joint IRAD project with Rockwell NAA. The panel was resin transfer molded using Shell 9405 resin. Figure 15 illustrates a 9 inch by 48 inch sine wave spar braided for Boeing Military Aircraft using Amoco T-650-42 graphite fiber and Radel X comingled thermoplastic. The photograph was taken after initial consolidation processing.

**TABLE 3**  
**REPRESENTATIVE 3-D BRAIDED COMPOSITE MECHANICAL PROPERTIES**

| <u>ORIENTATION</u> | <u>LOADING<br/>CONDITION</u> | <u>PROPERTY</u>     | <u>G-40</u> | <u>RTM*</u> | <u>FIBER</u> | <u>PRESS<br/>MOLD**</u> |
|--------------------|------------------------------|---------------------|-------------|-------------|--------------|-------------------------|
| 0/90°              | Tension                      | E <sub>x</sub> msi  | 9.2         | 7.5         | 14.2         | <u>AS-4</u>             |
|                    |                              | S <sub>x</sub> ksi  | 70.2        | 59.7        | 70.3         |                         |
|                    | Compression                  | E <sub>x</sub> msi  | 7.2         | 7.3         | 11.8         |                         |
|                    |                              | S <sub>x</sub> msi  | 35.0        | 35.1        | 32.4         |                         |
|                    | Shear                        | G <sub>xy</sub> msi | 0.6         | 1.1         | 0.4          |                         |
| ±45°               | Tension                      | S <sub>xy</sub> ksi | 9.0         | 13.1        | 12.4         | <u>UHM</u>              |
|                    |                              | E <sub>x</sub> msi  | 3.1         | 2.4         | 2.6          |                         |
|                    | Compression                  | S <sub>x</sub> ksi  | 35.8        | 21.5        | 11.5         |                         |
|                    |                              | E <sub>x</sub> msi  | 3.7         | 1.3         | 1.4          |                         |
|                    | Tension                      | S <sub>x</sub> msi  | 21.1        | 20.2        | 17.4         |                         |
| 0±60°              | Tension                      | E <sub>x</sub> msi  |             |             | 6.4          | <u>AS-4</u>             |
|                    |                              | E <sub>x</sub> msi  |             |             | 5.2          |                         |
|                    |                              | E <sub>x</sub> msi  |             |             | 58.4         |                         |
|                    |                              | S <sub>x</sub> ksi  |             |             | 35.0         |                         |
|                    |                              | S <sub>y</sub> ksi  |             |             | 5.7          |                         |
|                    | Compression                  | E <sub>x</sub> msi  |             |             | 5.2          |                         |
|                    |                              | E <sub>y</sub> msi  |             |             | 50.8         |                         |
|                    |                              | S <sub>x</sub> ksi  |             |             | 27.0         |                         |
|                    | Shear                        | S <sub>y</sub> ksi  |             |             | 2.0          |                         |
|                    |                              | G <sub>xy</sub> msi |             |             | 23.1         |                         |
|                    | Poisson's Ratio              | S <sub>xy</sub> ksi |             |             | 0.33         |                         |
|                    |                              | ν <sub>xy</sub>     |             |             | 0.30         |                         |
|                    |                              | ν <sub>yx</sub>     |             |             |              |                         |

\* Average fiber volume of 48% using DOW TACTIX 123/H91 epoxy resin.

\*\* Average fiber volume of 56% using Shell 9405/9470 epoxy resin.

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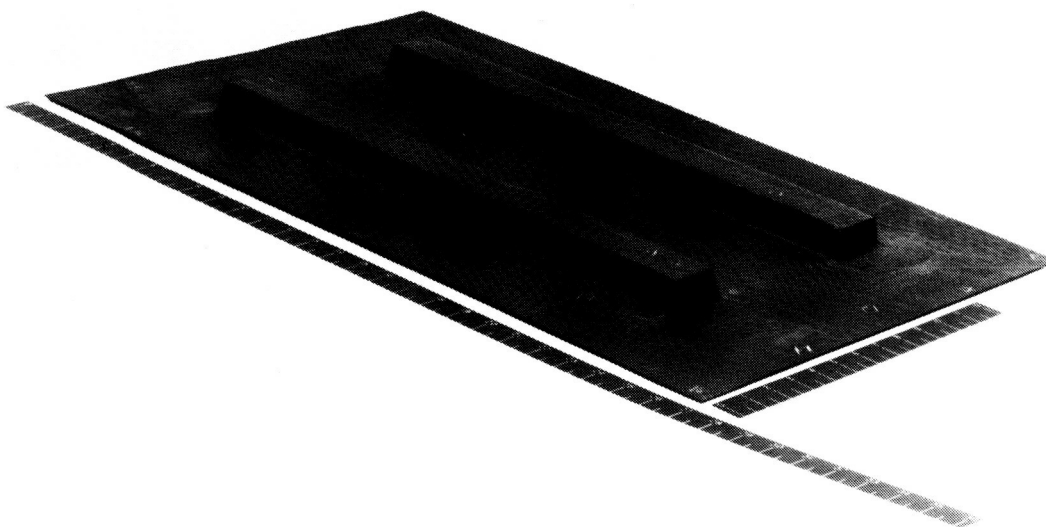


Figure 12

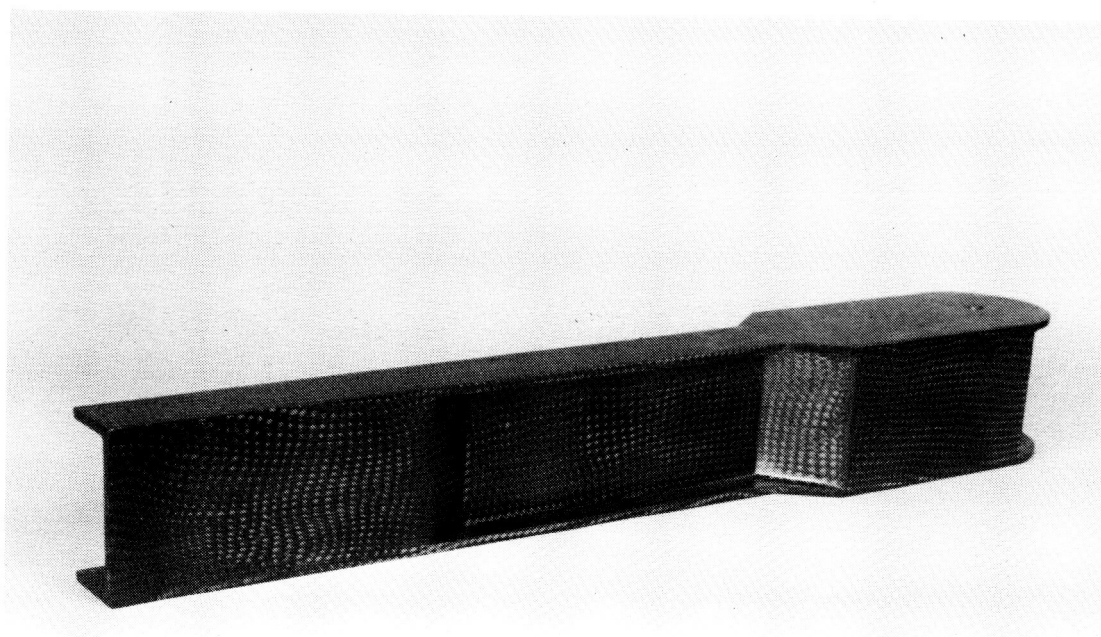


Figure 13

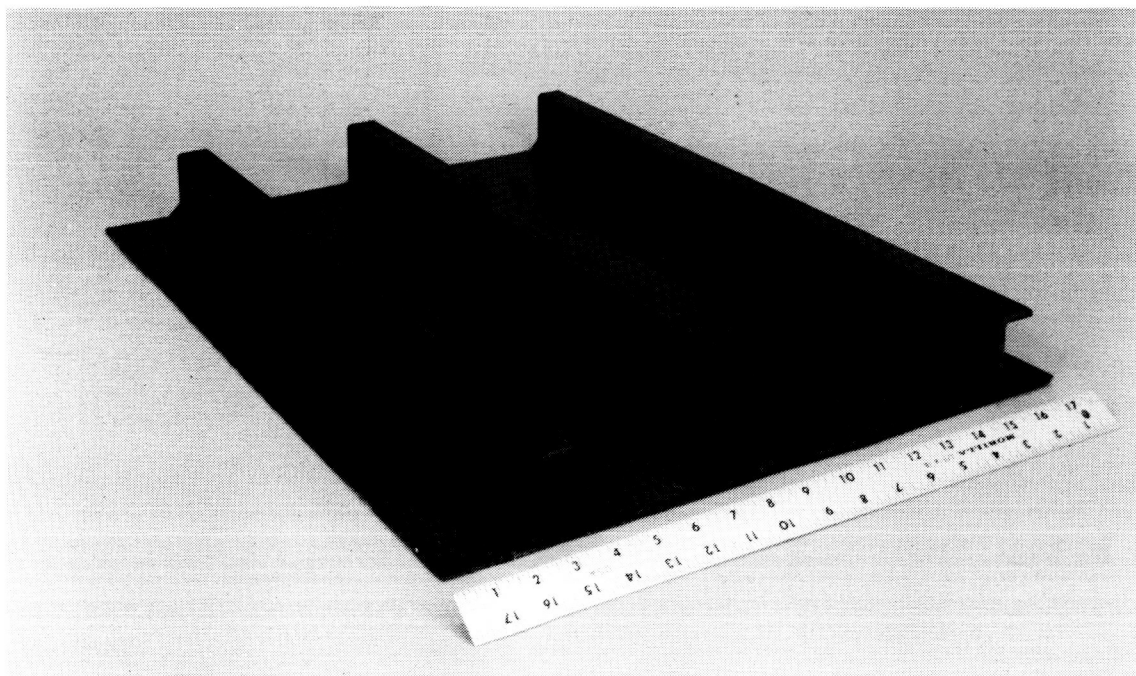


Figure 14

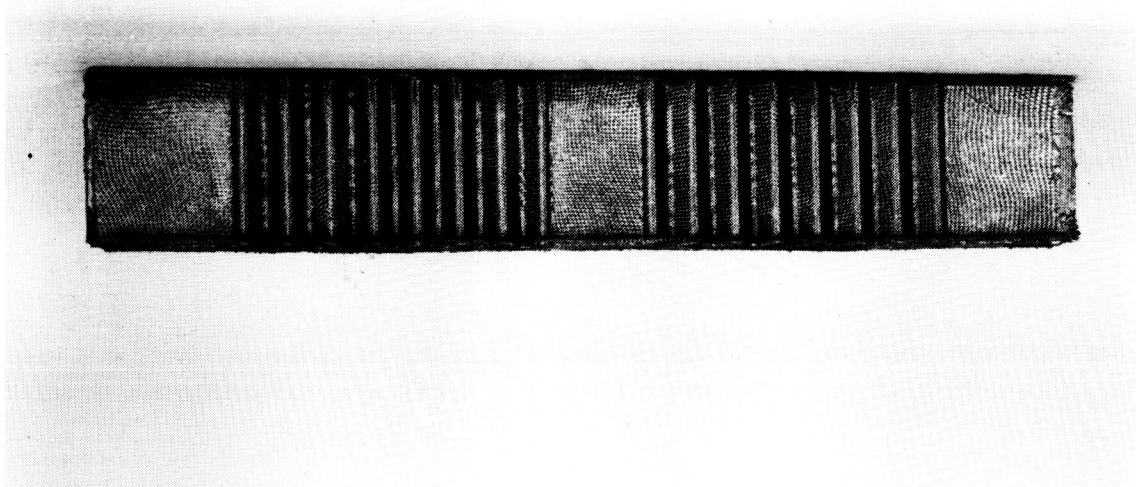


Figure 15

## CONCLUSIONS

This paper describes the performance of 3-D braided composites with respect to aircraft structural requirements. ARC's 3-D material possesses a high tolerance to impact damage. Compared to laminated composites, the damaged area is 52% lower for the same impact energy. In addition, the compressive strength is not appreciably reduced in the presence of damage whereas the compressive strength of a lamination is reduced by 40% of its original value. Reduced sensitivity to structural discontinuities, such as holes or cut-outs, is shown by the results of comparative open hole tension and compression tests. The load carrying capability of laminates was reduced by up to 50% while the 3-D braided composites exhibited little strength reduction. Finally, rib stiffeners integrally braided to a skin, demonstrated three-to eight-times the pull-off strength of bonded ribs.

Application of 3-D braided composites to aircraft structures should be guided by the same principles used in the selection of any material system. The material selection must be tailored to the structural requirements consistent with the advantages and disadvantages of the material under consideration. In some cases, notably the compressive strength of undamaged specimens, 3-D braided composites do not provide the same level of performance as laminates. This is because the gain in shear strength comes at the expense of axial performance.

It follows that the best areas for the application of 3-D braids are those components and locations where resistance to foreign object damage, insensitivity to cut-outs, or superior shear strength are required. In these areas, slightly lower initial tensile and compressive strengths are compensated for by the residual strength of the 3-D architecture.

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**Thermoplastic Pultrusion for  
Future Aerospace Application**

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McDonnell Douglas Missile System Company

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## ACT/ICAPS - Thermoplastic Composite Activities

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## SUMMARY

McDonnell Aircraft Company (MCAIR) is teamed with Douglas Aircraft Company (DAC) under NASA's Advanced Composites Technology (ACT) initiative in a program entitled Innovative Composite Aircraft Primary Structures (ICAPS). Efforts at MCAIR have focused on the use of thermoplastic composite materials in the development of structural details associated with an advanced fighter fuselage section with applicability to transport design.

Based on innovative design/manufacturing concepts for the fuselage section primary structure, elements were designed, fabricated and structurally tested. These elements focused on key issues such as thick composite lugs and low cost forming of fastenerless, stiffener/moldline concepts. Manufacturing techniques included autoclave consolidation, single diaphragm co-consolidation (SDCC) and roll-forming.



## INTRODUCTION

A major obstacle to widespread use of high performance composites in primary aircraft structures is the high cost of manufacture and assembly. Under NASA's ACT program, McDonnell Aircraft Company investigated cost-effective innovative techniques for the fabrication and joining of primary airframe structure using thermoplastic composite materials. While these developments were directed toward an Advanced Short Take-off or Vertical Landing (ASTOVL) aircraft, they are equally applicable to commercial vehicle structure. Efforts in these programs were conducted in design and structural mechanics development, manufacturing concepts development, and structural testing. Due to NASA program redirection, efforts were curtailed at the element level. This report summarizes progress through element verification. The elements addressed key design issues associated with the fuselage section; fastenerless frame attachment concepts, thick composite lugs, and rolled-formed stiffeners.

Structural verification testing was performed on the element specimens. Pull-off strength tests at ambient and elevated-temperature-wet conditions were conducted on the fastenerless frame concepts. To assess analytical developments, thick lug tests were performed under ambient conditions for two lug geometries and three laminate configurations.

## GENERIC FUSELAGE SECTION

The advanced aircraft system selected for the fighter development effort was the Model 4629 ASTOVL design developed by MCAIR under the NASA-Ames sponsored U.S./U.K. ASTOVL Technology Development program. Based on representative fuselage cross-sections of the Model 4629 aircraft, a generic center fuselage structure, Figure 1, was developed as the primary structure demonstration component. While the fuselage structure contains design features particular to advanced ASTOVL aircraft, cost-effective fabrication techniques and innovative design concepts developed in this program demonstrated technology related to all emerging aircraft systems.

## MATERIAL AND PROCESS SELECTIONS

### Material

The material chosen was based on temperature requirements, solvent resistance, component design, manufacturing approaches, and processing ease. Due to the 255°F design requirement, the baseline thermoplastic resin system selected was ITX (intermediate temperature crystalline), which has service capability to 300°F. ITX processing temperature and pressure ranges are 700°-750°F and 100-150 psi, respectively, as such processing characteristics are similar to ICI's APC-2 (PEEK) system. Amorphous systems were not considered due to their poor resistance to solvents such as jet fuel.

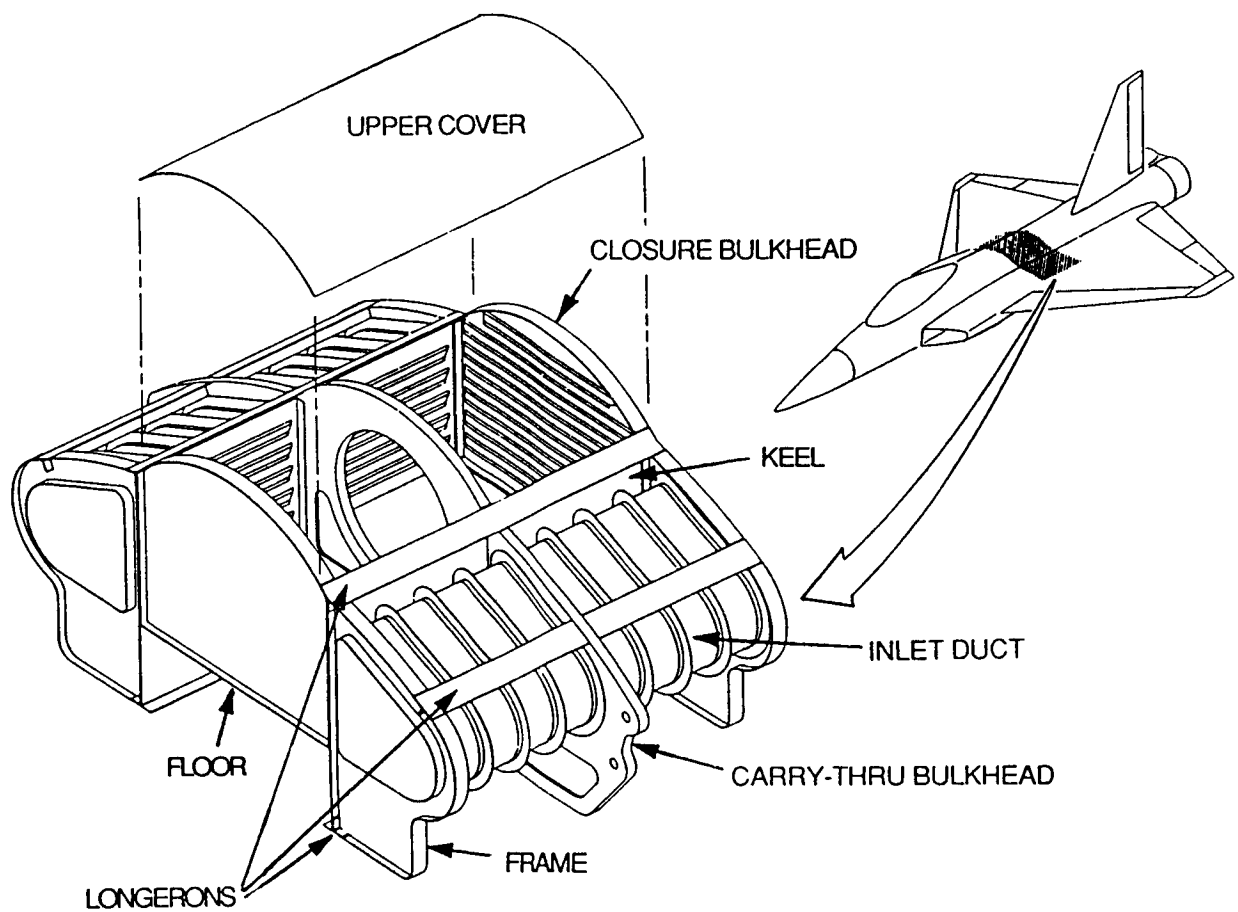


Figure 1 Generic Fuselage Section Offered a Full Range of Design and Manufacturing Challenges

The fiber selected was IM7, an intermediate modulus fiber produced by Hercules. In addition, AS4/APC-2 was selected for early forming studies due to immediate availability and to verify analytical predictions for thick composite lugs.

### Processes

Manufacturing processes were selected using a concurrent engineering approach. Processes were rated based on innovativeness, cost, risk, supportability, survivability, and weight. Two manufacturing techniques, fiber placement and single diaphragm/coconsolidation (SDCC), were determined to be potential low cost fabrication techniques (Reference 1).

Fiber placement (FP), one of the more promising methods of fabrication, is the in-situ consolidation of individual material layers using pressure and heat at the point of contact. This procedure eliminates the autoclave requirement and automates the material deposition process reducing significant cost elements in a typical composite production environment. Work using this approach area was discontinued due to program redirection.

Single diaphragm/coconsolidation (SDCC) simultaneously consolidates the hat stiffened inner skin plies with the outer skin plies while coconsolidating the two yielding a high quality interface and reducing the number of process steps from three to one. This approach was used to fabricate representative fastenerless frame attachment concepts developed in this program.

### SDCC Manufacturing Concept

The SDCC concept is unique in that there is but one diaphragm, and the IML pan and OML skin are coconsolidated during the diaphragm forming process. The SDCC tooling concept is illustrated in Figure 2. A vacuum frame is utilized to hold the IML ply pack prior to heat-up and pressurization. The greatest risk in diaphragm forming over hat mandrels is the chance for bridging. To reduce this risk, hat spacing is minimized to increase ply surface area between mandrels. The increased surface area increases the forces exerted to form the ply pack and prevent bridging. In addition, the mandrels are fabricated with a slight radius, Figure 3. The gap between the mandrel, OML skin and IML pan are filled with a predetermined amount of unidirectional tow. This fillet area has the highest probability for bridging; however, with the unidirectional fillet, the pressure would be equally distributed to facilitate a quality consolidation.

A vacuum ring and a neat film layer aid in ply pack location. The IML ply pack is contained between the aluminum diaphragm and a layer of neat film. A vacuum ring surrounds the IML ply pack and vacuum draws the aluminum diaphragm to the upper surface of the IML ply pack, and the neat film to the lower surface of the IML ply pack. The IML ply pack is then positioned above the tool prior to application of heat and pressure. The neat film is coconsolidated between the IML and OML ply packs during the press operation. This permits accurate location of the IML ply pack and aids in the prevention of wrinkles.

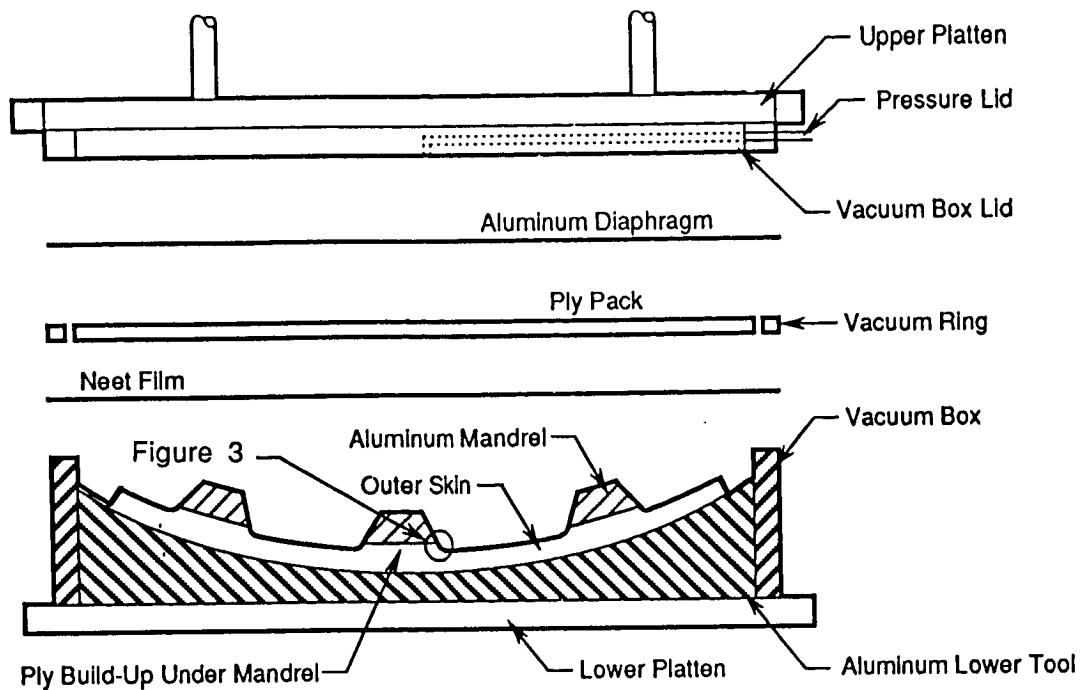


Figure 2 Single Diaphragm Coconsolidation (SDCC) Concept  
Offered Potential for Low Cost Manufacturing

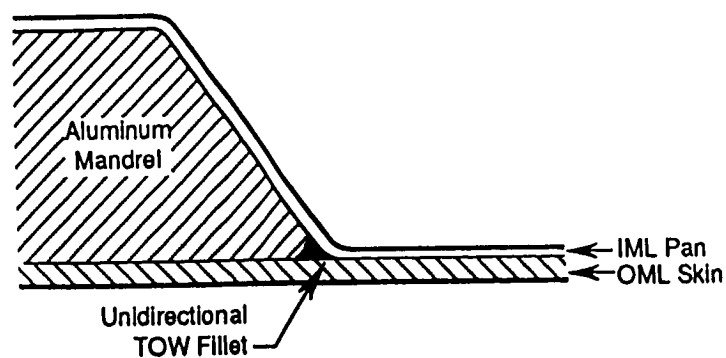


Figure 3 Unidirectional Tow Used in Fillet Area to Assure Part Quality

## THERMOPLASTIC FUSELAGE ELEMENT CONCEPTS

As a first phase to the building block approach to the production of a full-scale fighter fuselage section, element components were selected for design, analysis, and structural validation. Element selections were sought which would address key areas of the innovative composite design including structural design, analysis, and manufacturing development. In addition, the elements were to be representative of key components within the future phases of the subcomponent development.

Two structural areas of particular interest in the fuselage structure were identified by the Design/Manufacturing Integration (D/MI) team. Stiffener to skin joints were selected based upon:

- o anticipated increases in performance through optimal design for pull-off and shear-transfer loading
- o decreased costs through manufacturing innovations such as the SDCC approach
- o potential for LO applications, reduced weight, and reduced assembly due to the fastenerless design associated with SDCC
- o current MCAIR developments for the analysis of stiffener pull-off strengths and comparative test data

Thick composite lugs were identified as the second element for evaluation based upon:

- o anticipated increases in performance through optimal design for pin-bending effects and through-the-thickness loading effects
- o decreased costs through the validation of water-jet-cutting of thick composites for initial and final trim
- o current developments in thick composite analysis which could be enhanced to provide a useful design/analysis tool for the evaluation of highly loaded and out-of-plane loading of thick composites

### Frame Elements

Frame Element Design - A fastenerless moldline Y-frame attachment design was selected due to its applicability to the diaphragm forming coconsolidation technique. The design was also anticipated to generate substantially greater strengths for pull-off and shear transfer due to the increased interlaminar shear and tension strengths associated with semicrystalline thermoplastic composites. The design therefore exploited both the mechanical and manufacturing strengths of the thermoplastic material system.

The Y-frame layup consisted of identical outer-moldline (OML) and Y-section (IML) laminates [+45/90/0/90/+45]. The inner and outer skin was to be coconsolidated during the diaphragm forming process. The only design variable was the attachment angle of the two Y-frame legs. This geometric parameter controls the ratio of shear to flat-wise tension loading at the frame/skin interface. In order to establish a better understanding of both the material allowables, failure modes, and failure locations, two (2) geometries were chosen. A 45° and 60° angle of incidence was established based upon initial parametric evaluations. Although initial analysis was unable to show a distinct difference in failure mode for these two geometries it did identify differences in the failures loads associated with each. The finalized designs incorporating the SDCC manufacturing method and the differing leg angles are shown in Figure 4.

In order to establish a baseline comparison, blade elements were also designed and analyzed. The blade configuration, Figure 5, incorporates back-to-back angle laminates which are identical in layup to the Y-section legs. The OML skin was chosen to be equivalent to the "bay skin" used in the Y-section, [+45/90/0/90/+45]s. This design yields a substantially stiffer skin section beneath the frame attachment, but was chosen in order to maintain the critical stacking sequence in the corner radius and provide an identical skin laminate in the area of applied constraints for the testing procedure.

SDCC Y-Frame Elements Fabrication - An SDCC element verification tool, Figure 6, was developed which incorporated either two hat mandrels or a single triangular mandrel to simulate fabrication of stiffener and frame designs, respectively. The entire tool was located in a vacuum forming box to facilitate the SDCC manufacturing process.

The verification tool was first utilized as a parallel hat-section tool to verify stiffener fabrication. The hat stiffener mandrels, located by pins, float on the unconsolidated skin. The inner skin was then formed over the mandrels in a press operation. A pinning arrangement allows the mandrels to float during consolidation. These pins permit the consolidation force to be transferred through the mandrels to the part to insure a quality consolidation. The aluminum tools were readily extracted following forming.

Due to program redirection, hat-section forming trials were discontinued and attention concentrated on forming of the Y-frame elements. Lessons learned from the hat-section trials were incorporated into this effort. To minimize diaphragm rupture due to sharp corners, the tooling was modified by extending the mandrels to the ramps of the forming box, Figure 7. In addition, application of full pressure should take place after the plies have reached melt temperature instead of just before melt. This allows the matrix to reach its minimum viscosity before forming is initiated. Previously, pressure was introduced prior to melt temperature to encourage interply slippage between the upper and lower ply packs. Due to the bridging observed on the room temperature forming trials, it was decided that forming depth would be improved if the ply pack was at full melt temperature.

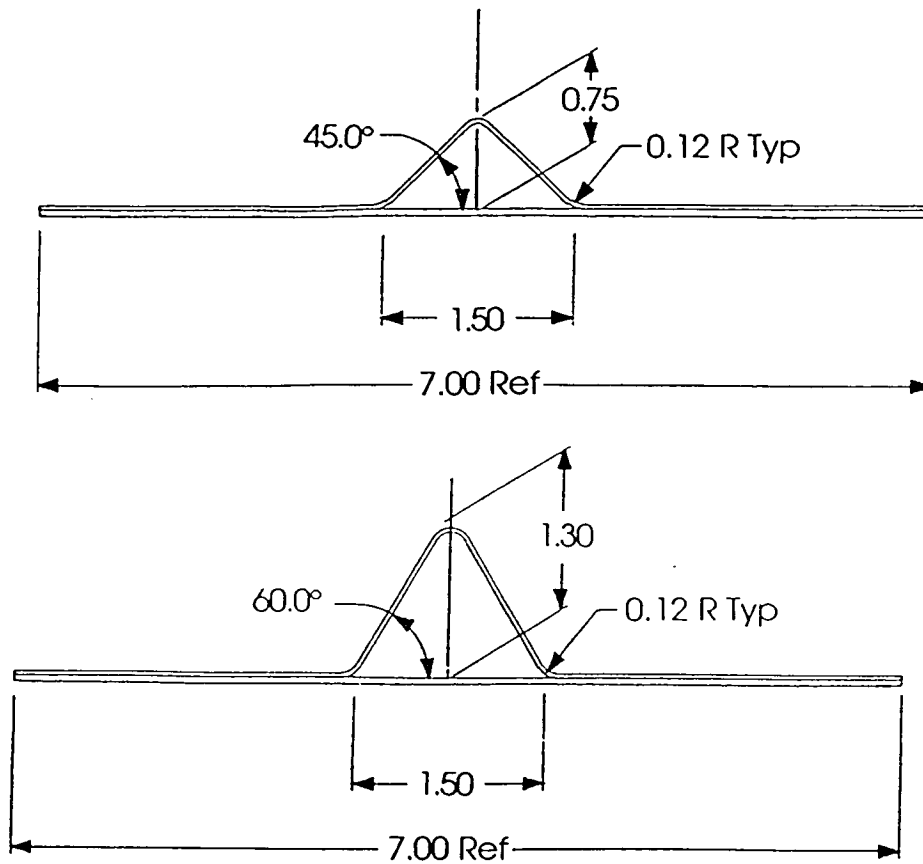


Figure 4 SDCC Y-Frame Element Designs Used to Determine Viability of Fastenerless Attachment Concepts

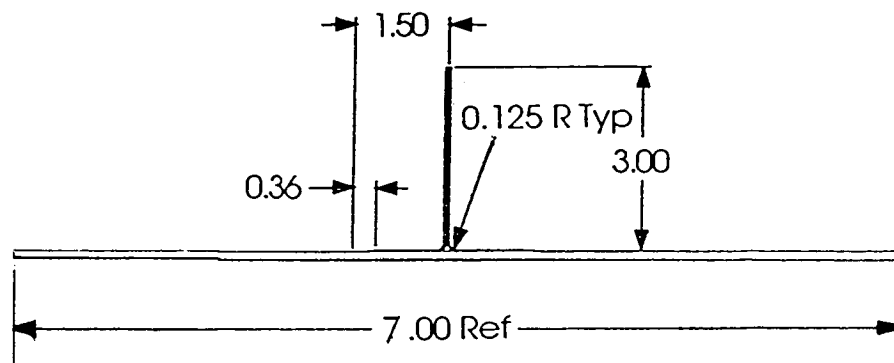


Figure 5 Blade-Frame Element Design Used for Baseline

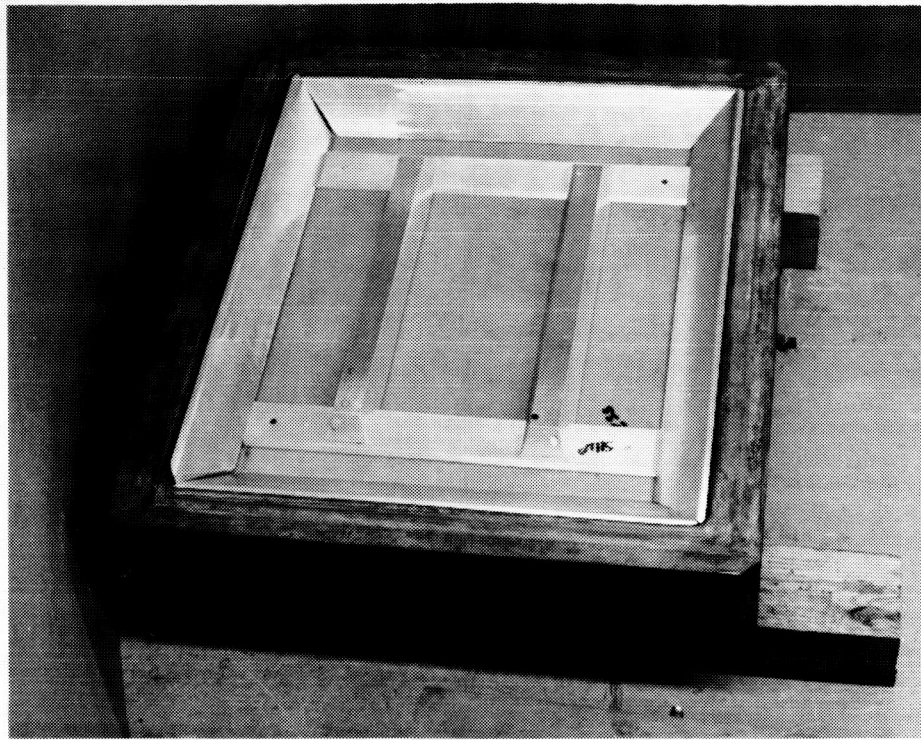


Figure 6 SDCC Element Verification Tool Incorporated Full Scale Design Features

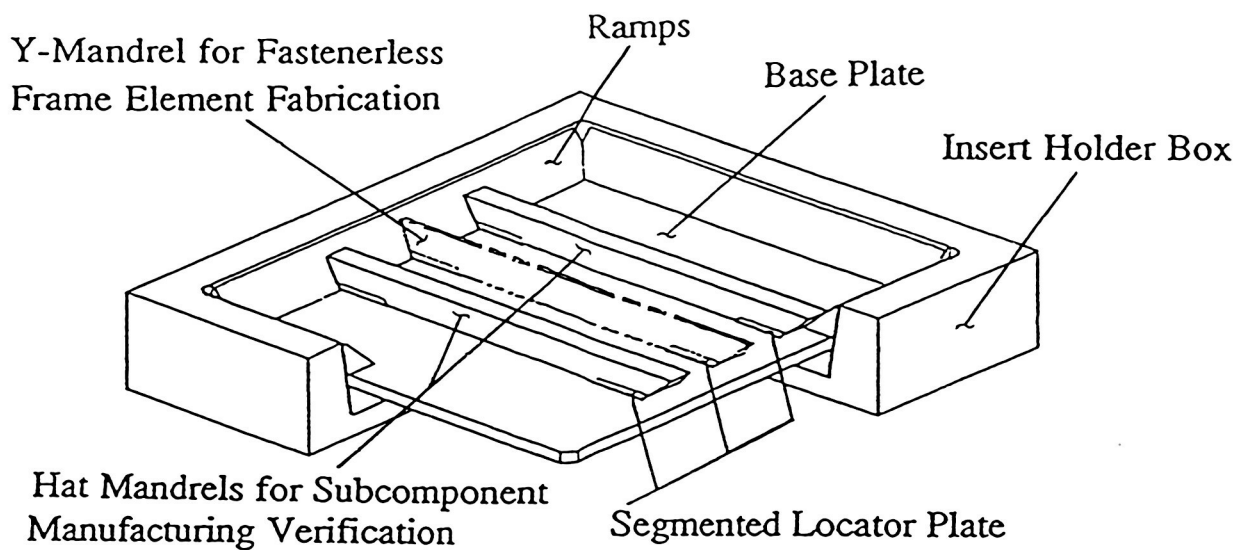


Figure 7 Modifications to SDCC Element Verification Tool Eliminated Early Diaphragm Ruptures (Segmented Locator Plates and Mandrel Extension)



Forming trials on the single "Y" configuration used one diaphragm to consolidate the upper ply pack with the lower plies. Initially, full pressure (150 psi) was applied after the melt temperature of the ITX was reached, but was maintained for only 5 minutes at which point the diaphragm ruptured. In spite of the short hold time the pressure was sufficient to fully consolidate the flat areas of the part and to form the material over the mandrel. The upper ply pack conformed to the mandrel surface, but diaphragm rupture caused the outer ply to lift and bridge across the mandrel/skin intersection. The other plies remained in the formed condition, nesting closely to the mandrel and showed an excellent definition at the interface between stiffener web and lower skin.

Forming was next done below melt temperature because of anticipated problems where the two ply packs met each other beyond the stiffener area. For the next trial full melt temperature was achieved before pressurizing. The upper ply pack was also widened so it extended out to the ramps in all directions. This change required notches to be cut along the edge of the ply pack so it wouldn't buckle and rupture the diaphragm. Kapton tape was used to cover the notches for additional protection.

Forming trials were performed with the above changes and the diaphragm survived fairly well up through 150 psi. Since the plies were well melted by this time, relatively good consolidation was achieved between the upper and lower packs. Rupture occurred along the edge of the mandrel in a notch location that allowed the film to over elongate. A large percent of the plies remained formed to the mandrel surface along its base. Only one ply lifted and bridged away from the radius area of the formed plies, Figure 8. The inside of the stiffener shape revealed very good contact between the plies being formed and the base of the mandrel even with loss of the diaphragm. Photomicrographs of the area revealed the upper plies dragged the lower plies in toward the mandrel and formed wrinkles in the lower skin.

In an attempt to alleviate dragging of the base ply pack, the upper ply of the base pack was extended to run under the ramp areas of the tool. This change would maintain pressure on the top ply to allow slippage of the two ply packs without wrinkling. Also, a fiber glass cloth (picture frame) was placed around the ramp areas and over the mandrel to cover any areas that could potentially allow the diaphragm to rupture. An additional change to the process was to initiate application of the forming pressure at 600°F. The rationale was to apply pressure below melt-temperature of the material to allow ply slippage prior to a viscosity change. During this run the diaphragm ruptured in a gap between the ramp and forming box causing incomplete forming of the element. However, improved ply slippage was noted due to the reduced temperature.

Due to the frequency of rupture of the UPILEX diaphragms, an aluminum (SUPRAL) diaphragm was selected for further trials. The aluminum diaphragm offered greater elongation capabilities not only at processing temperatures, but also at temperatures below the melt temperature of the PEEK resin.

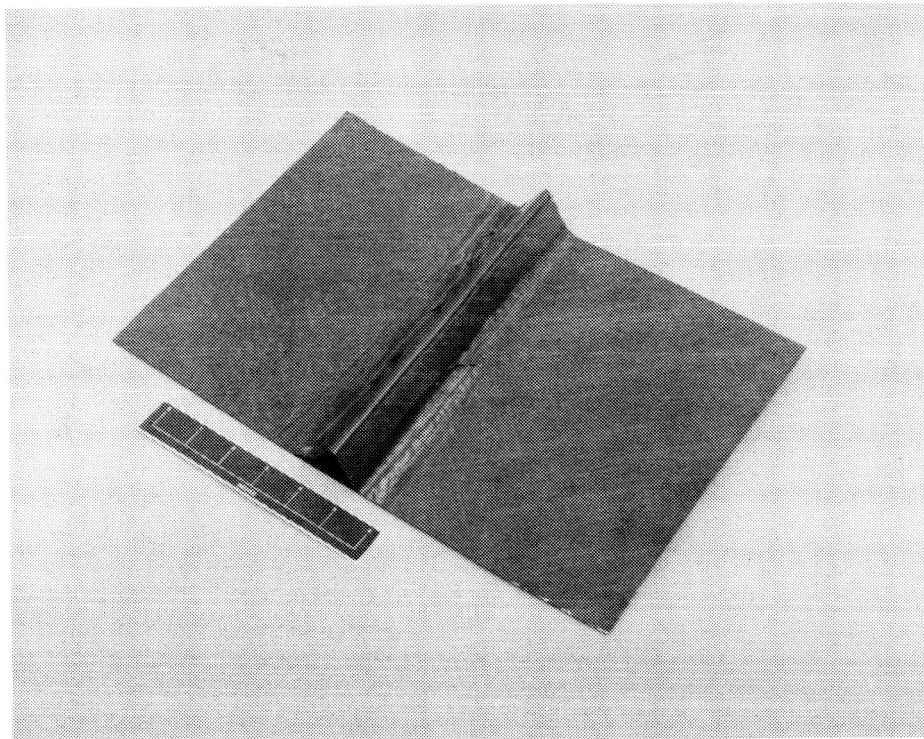


Figure 8 Initial SDCC Y-Frame Element Experienced Bridging in Radius

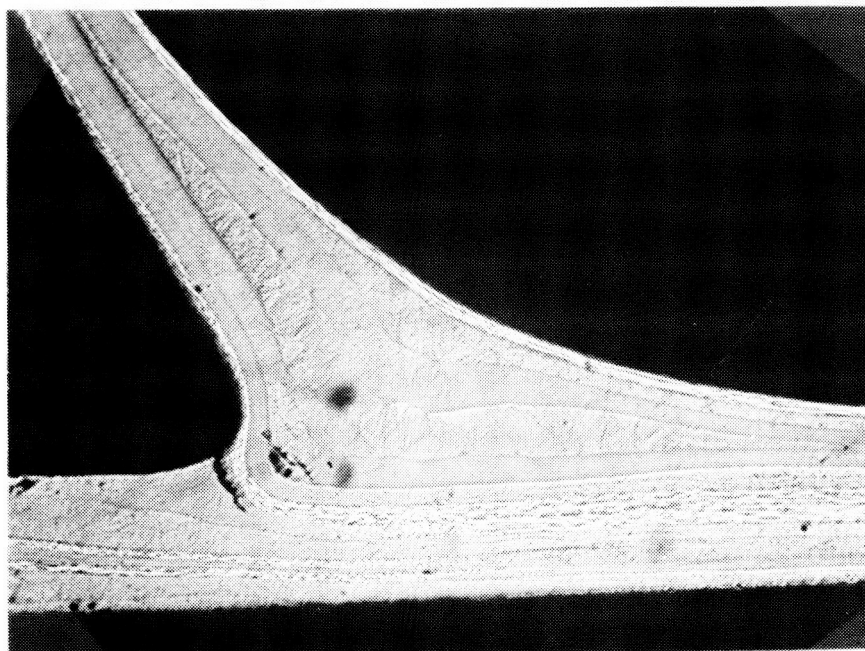


Figure 9 Ply Wrinkling Occurred in SDCC Y-Frame Corner Radius During One-Step Forming

During the first run with an aluminum diaphragm, the pressure was applied at 550°F. Applying the pressure at this low temperature allowed for additional ply slippage prior a viscosity change of the resin. During this fabrication attempt, the top ply of the lower ply pack was extended beneath the forming ramps in an attempt to "lock" the ply in place, thus avoiding wrinkles. After applying pressure (120 psi) at 550°F, the temperature was increased to 750°F and held for 30 minutes.

The result was a stiffened panel with good surface quality but bridging in the radius. NDE results revealed a porosity free part in the flat areas. However, photomicrographs revealed the lower ply pack wrinkled. Since the upper ply of the lower ply pack wrinkled, and the ends were contained beneath the forming ramps, the ply obviously split between the fibers of this outer 45° ply.

Following review of the results of the run, two changes to the manufacturing process were identified to alleviate the wrinkling problem in the next run. First a .003" layer of neat resin was applied to the bondline to serve as lubricant. The film would allow the two contacting plies to slip past each other and form in the radius area. A photomicrograph of the radius area following processing utilizing the neat resin is presented in Figure 9. As can be seen, radius cracking and bridging are present along with wrinkling in the lower ply pack. 0° plies were then added at the interface to allow the inner and outer ply packs to more readily slip past one another. The results revealed similar problems as noted with the neat resin.

To overcome friction between the two surfaces, it was decided that the upper ply pack should be driven into the radius areas prior to reaching the glass transition temperature. This would allow the plies to slip past one another prior to a viscosity change. To accomplish this, the upper ply pack was preconsolidated and then coconsolidated to the lower ply pack in a second operation. With this approach the risk level is significantly reduced while maintaining a major processing cost reduction.

The 45° and 60° Y-stiffened element panels were fabricated successfully using the two-step process as typified in Figures 10 and 11. The upper ply pack was preconsolidated utilizing the diaphragm forming process. Following consolidation, the upper ply pack was coconsolidated to the lower ply pack. Although this is a change from the original SCDD concept, a cost savings is still realized by reducing the process to two steps from a traditional three-step process.

The 45° and 50° Y-stiffener panels were nondestructively evaluated. The C-scan results on the 60° panel showed a quality part free of voids. The 45° Y-stiffened panel, however, revealed slight porosity under the mandrel. This was due to a piece of sheet metal slipping beneath the mandrel during consolidation. The sheet metal was modified and another 45° panel fabricated with excellent results from NDE.

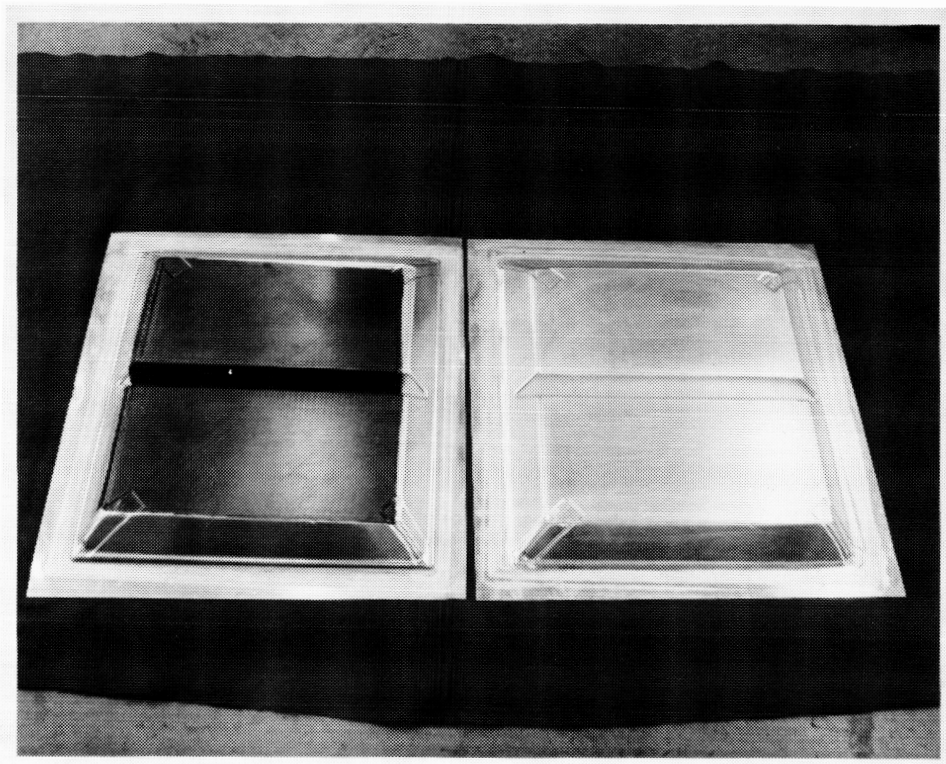


Figure 10 Aluminum Diaphragms and Two-Step Forming Provided Quality Y-Frame Elements

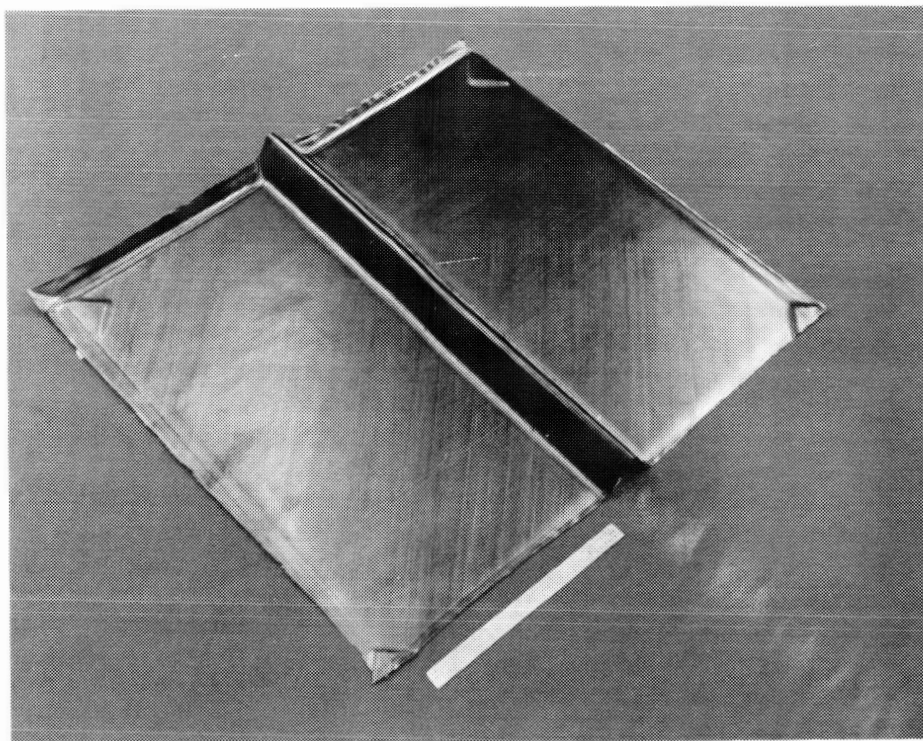


Figure 11 Typical Two-Step Diaphragm Formed 60° Y-Frame Element

The dual-step SDCC process reflects a significant cost reduction compared to conventional diaphragm forming of integrally stiffened skin structure which involves a three-step approach; forming of the inner corrugated skin, forming of the outer skin, and subsequent coconsolidation. In addition, the three-step process involves the use of an additional diaphragm for outer skin forming.

Our original intent was to develop a single-step SDCC process. Our attempts were unsuccessful due primarily to dragging of outer moldline skin plies into corner radii of the corrugations during forming of the inner moldline skin. The dual-step process was then adopted to insure timely fabrication of the element test specimens.

Blade Frame Elements Fabrication - Using two aluminum block details a blade panel was hand laminated by bending and edge tacking each of seven plies with a soldering iron, Figure 12. The fillet was filled with thin strips (.30" to .90") of ITX unidirectional tape using a sharp cone tip on the soldering iron. A flat skin was preconsolidated and a strip grit blasted across the center where the blade attached, Figure 13. The two angles with fillet in place were inverted onto this skin, Figure 14, and vacuum bagged to a project plate. There was a released UPILEX film between the angle plies (web) and the aluminum details. Upon consolidation, at 750°F and 100 psi, the part did not show acceptable C-scan results. The web area had many depressions in it that appeared to be oriented along the second ply down from the surface, i.e., normal to the surface ply fiber direction.

Outgassing from the release coated UPILEX and the lack of ears on the vacuum bag at the base (which may have prevented sliding of the blocks) were identified as probable causes for the poor consolidation. As such, a second blade was fabricated with no UPILEX on the tool details and with extensive ear folds in the vacuum bag. Nondestructive inspection of the second blade revealed porosity in the radius areas. Although the part quality was improved over the first blade, it was not the level desired. One of the tool details had rotated during consolidation causing poor consolidation in the web and radius areas.

The consolidation tools were then modified to permit a positive control of the details. A trimetric view of the modification is shown in Figure 15. Keyways were milled into the ends of the web details and fit to guides in the end plates. This modification maintained the movement of the detail in the direction desired thus maintaining constant and equal pressure across the part surfaces. In addition, an upper slotted plate maintained minimum differential vertical displacement between the tooling blocks for the back-to-back L-section which comprises the T-section.

The modified tooling concept is a positive drive concept designed to maintain cap and web thickness while providing adequate pressure on the part. Conventional forming tools utilize mechanical stops to provide constant web and cap thickness. However, mechanical stops can contact the project plate prematurely reducing pressure on the part and ultimately causing porosity.



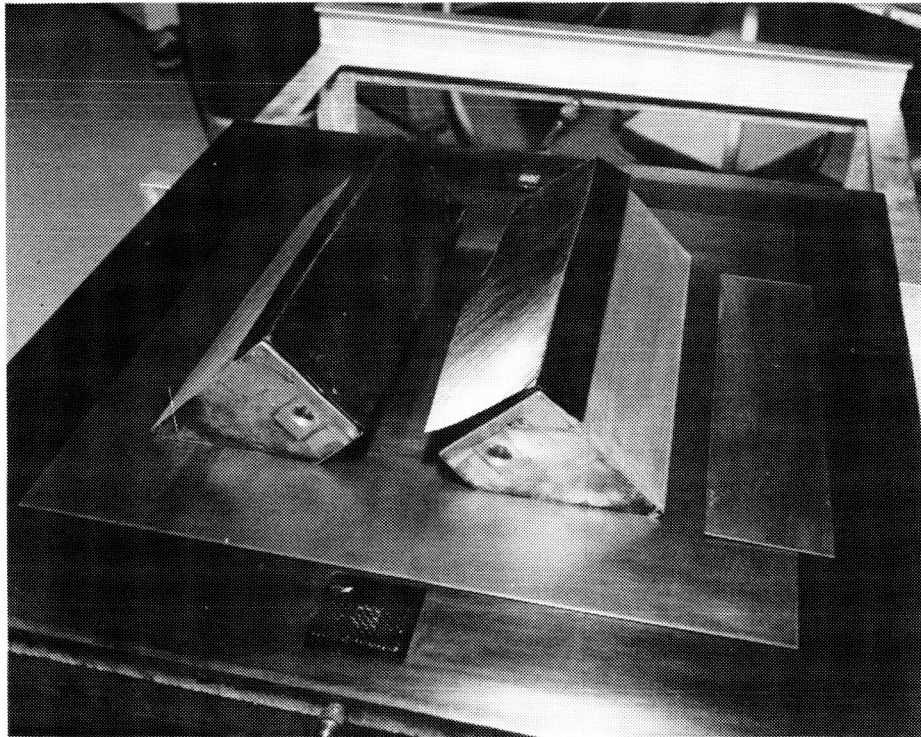


Figure 12 Blade L-Section Plies were Formed on Tooling Blocks to Form Web

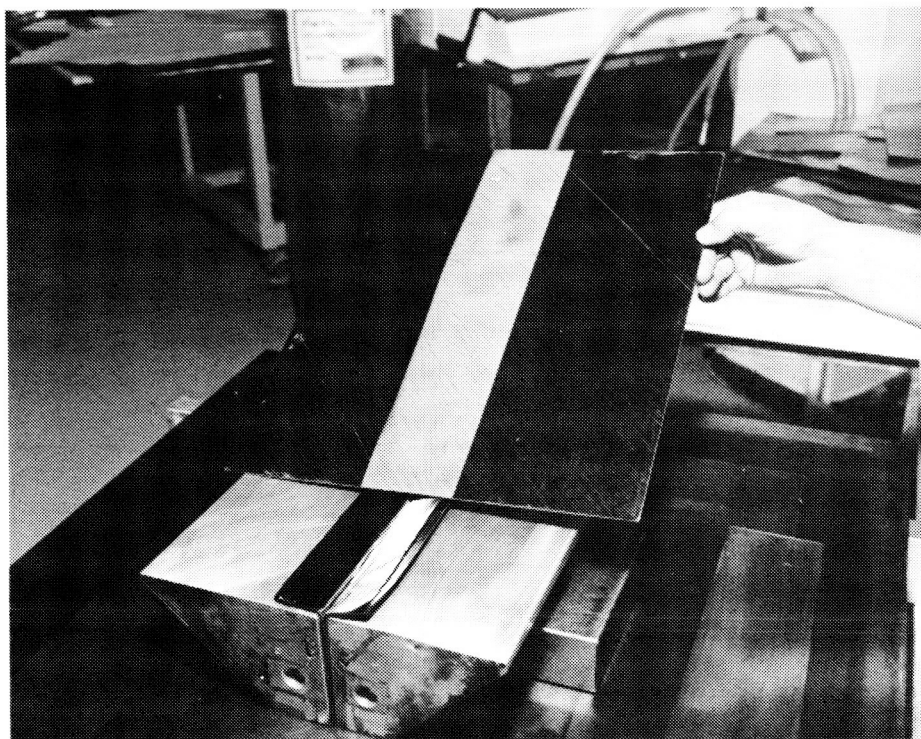


Figure 13 Base Plies were Consolidated Separately and Grit Blasted Prior to Assembly

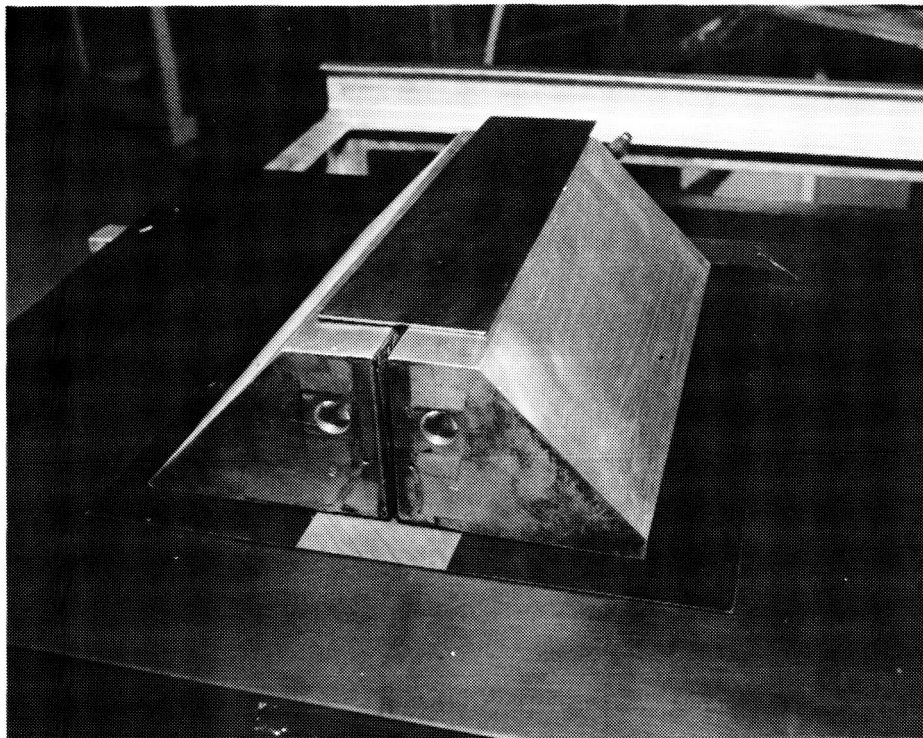


Figure 14 Blade Laminate/Tooling Assembly Prior to Autoclave Consolidation

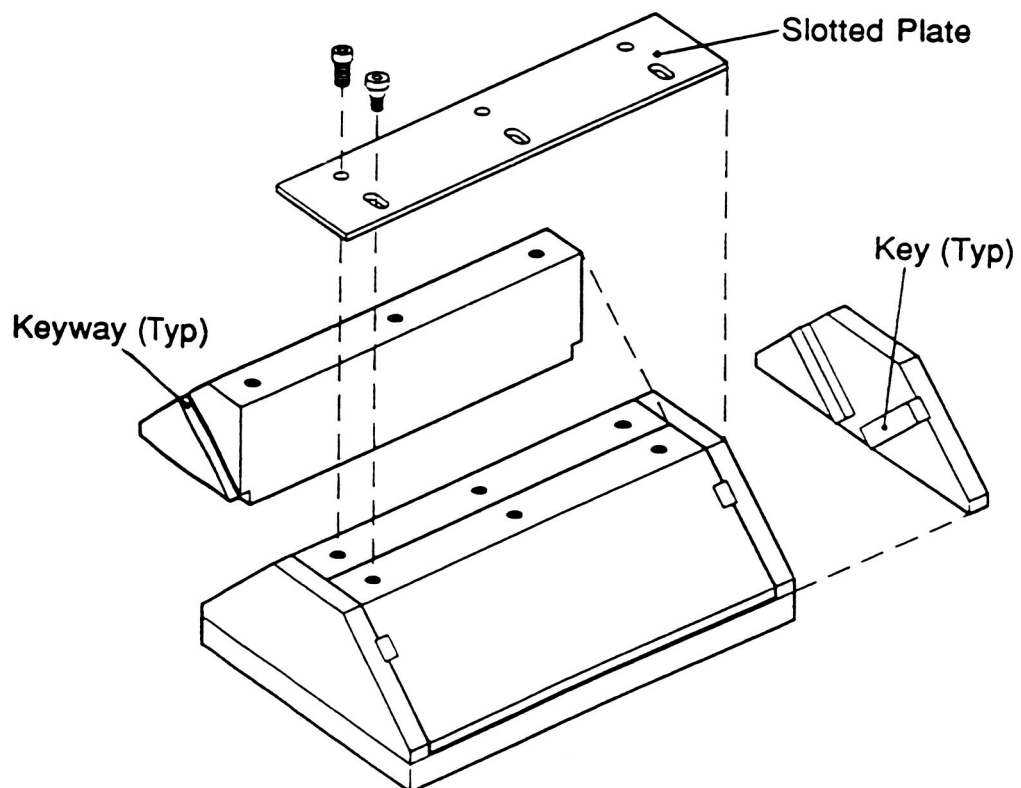


Figure 15 Modification to Blade Element Tooling Provided Necessary Pressure on Web and Radius

Using the same lay-up approach previously described and the positive drive tooling concept, a quality part was achieved on the first attempt, Figure 16. However, due to amount of fillet material used, a slight void was noted in a photomicrograph of the radius area. Following review of the photomicrograph, it was concluded that excessive  $0^\circ$  tow was being used in the fillet, and that the dwell time should be increased. During a second run, the two changes were incorporated in the consolidation process. The element was then C-scanned revealing a quality consolidation. Photomicrographs show total consolidation in the fillet area as well as the cap and web. A thickness check of the cap and web show only a .001" variation. This element panel was cut into specimens for pull-off testing, Figure 17.

A blade panel previously consolidated with the original tooling concept and exhibiting unacceptable levels of porosity was reconsolidated using the positive drive tooling. Both C-scan and photomicrographic inspections revealed a quality part. This blade element was also machined into element specimens in order to investigate the effects of reconsolidation. It is worthy to note that this latter scenario demonstrates potential cost savings associated with the ability to reconsolidate thermoplastic composites.

Frame Element Tests - Frame element testing was conducted on the fastenerless frame attachment designs which were selected for subcomponent and full-scale development. In addition, testing was conducted on an induction welded frame concept in order to initially assess the feasibility of this joining method for later use under the full-scale development.

Nondestructive Evaluation (NDE) revealed some minor porosity and defects in the  $45^\circ$  Y-specimen panel. An additional panel was fabricated to provide specimens. With the exception of induction welded Y-frame specimens, all coconsolidated specimens were determined to be of good quality. Difficulty in the examination of the corner radii was experienced for all specimens. Based upon available C-scan data and examination of the specimens after final trimming all specimens were accepted.

A dimensional check was conducted to further verify the accuracy of fabrication of all specimens and to establish a database for later failure correlation. The results of the dimensional checks showed minor variations (less than 10% deviation from nominal) in thicknesses and specimen widths. The width dimension was utilized to normalize all reported load data to pounds per unit width.

All specimen configurations were subjected to two (2) testing conditions as shown in Figure 18. Room temperature dry (RTD) testing was conducted on specimens which underwent an initial weighing followed by exposure to  $250^\circ\text{F}$  until weight loss stabilized. Elevated temperature wet (ETW) test specimens experienced the same  $250^\circ\text{F}$  drying exposure followed by moisture conditioning at  $160^\circ\text{F}$  and 95% RH. Based on a time history of the moisture conditioning, an average equilibrium moisture content for the LM7/ITX frame element was found to be 0.34% by weight.



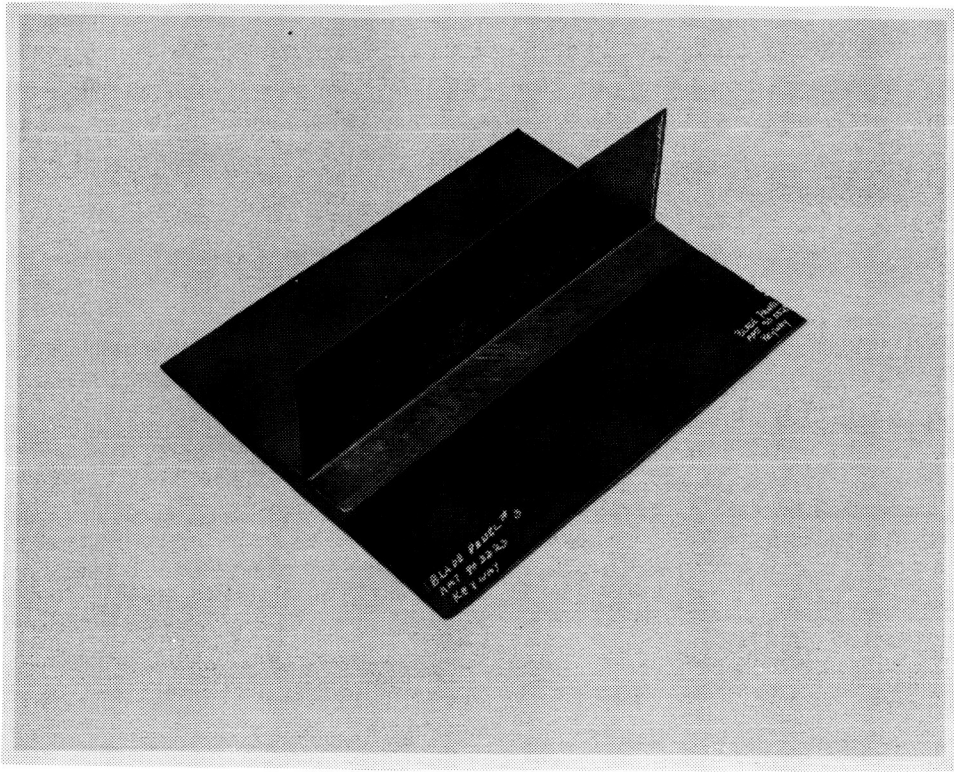


Figure 16 Typical Blade Frame Element

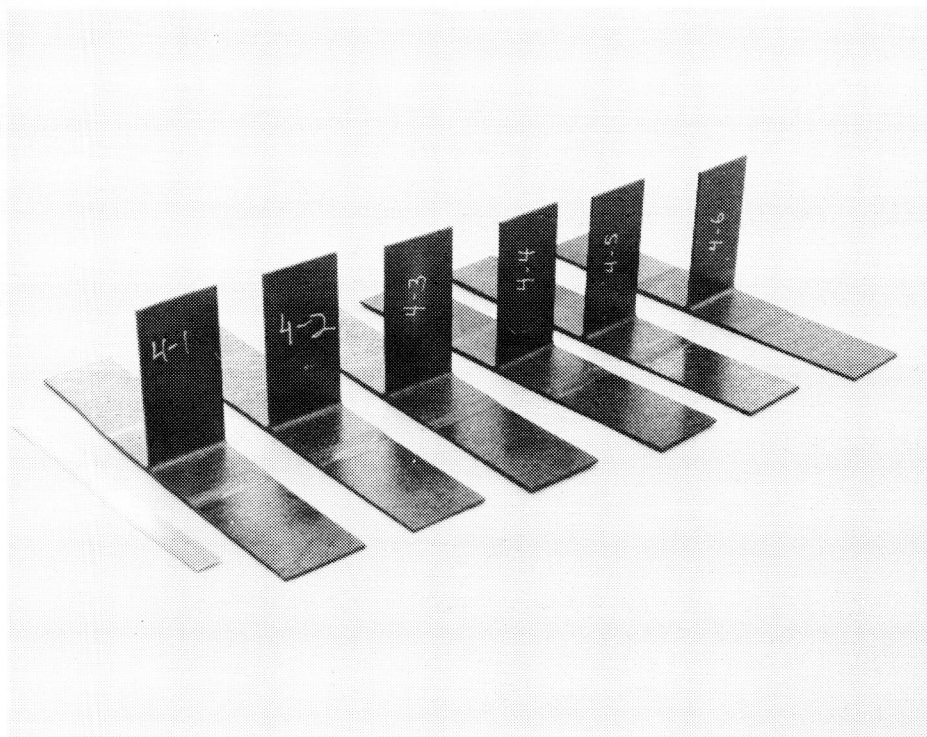


Figure 17 Blade Frame Element Panels were Machined into Pull-Off Specimens

| Frame Configuration | Fabrication Process | Number of Specimens at Test Environment |       |
|---------------------|---------------------|-----------------------------------------|-------|
|                     |                     | RTD*                                    | ETW** |
| Blade               | Co-Consolidated     | 3                                       | 3     |
| 45° Y-Frame         | Co-Consolidated     | 3                                       | 3     |
| 60° Y-Frame         | Co-Consolidated     | 3                                       | 3     |

**Figure 18 Fastenerless Frame Element Testing Matrix Used to Demonstrate Increases in Pull-Off Strengths**

The test setup for room temperature specimens is shown in Figure 19 for both the baseline blade and Y-frame specimens. Load introduction for the blade specimens was accomplished through direct gripping of the upstanding flange. A loading mandrel and clevis were utilized for the Y-frames. Initial testing showed substantial deflections due to the 5" span used between end clamps. In order to eliminate excessive deflections the test procedure was modified to provide a 3" span. The identical setup was used for the elevated temperature testing, with the test apparatus enclosed within a temperature control chamber. A thermocouple was utilized to ensure accurate control of the temperature to the required 250°F. A hold time of 5 minutes was utilized to ensure temperature uniformity for the part while reducing the risk of desorption associated with longer hold times.

Testing was carried out utilizing a displacement control rate of 0.1 in/min. Load versus deflection plots were obtained for each test condition. In addition to on-line recording of the load history, continuous visual inspection of the specimens was carried out during testing in order to establish initial failure modes and a correlation to the loading data. Videotaping of the initial room temperature tests provided a means of reviewing the test procedures (including load and displacement histories monitored on digital readouts). In addition, specimen edges were painted white prior to testing in order to provide contrast and enhance visual identification of failure initiation and location.

A summary of test results is presented in Figures 20 and 21. The following sections provide a detailed discussion of each of the specimen groups that were examined under element testing. A final overview and discussion of results follow these sections.

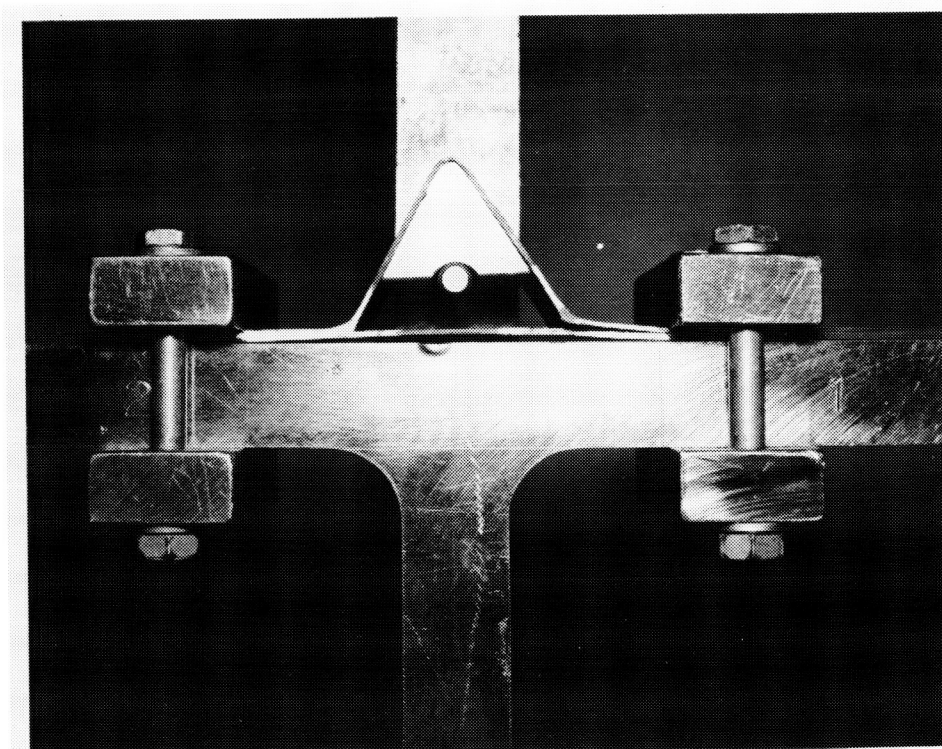
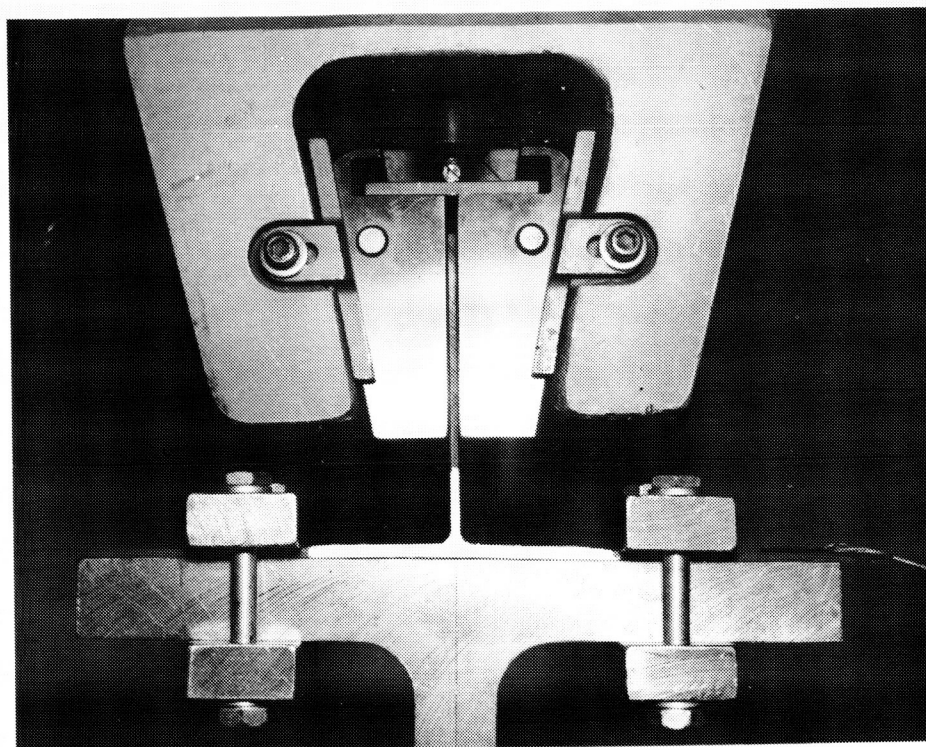


Figure 19 Typical Frame Element Pull-Off Test Set-Up

| Frame Configuration | Initial Failure |                 |                               | Final Failure |                 |                         |
|---------------------|-----------------|-----------------|-------------------------------|---------------|-----------------|-------------------------|
|                     | Load (lbs/in)   | Deflection (in) | Failure Mode                  | Load (lbs/in) | Deflection (in) | Failure Mode            |
| Blade               | 328             | 0.124           | ILT Failure* at Corner        | 315           | 0.178           | Interface Failure**     |
| 45° Y-Frame         | 554             | 0.148           | Skin Surface Failure at Clamp | 655           | 0.216           | ILT + Interface Failure |
| 60° Y-Frame         | 504             | 0.161           | Skin Surface Failure at Clamp | 574           | 0.247           | ILT + Interface Failure |

\* Interlaminar Tension Failure at Frame Corner

\*\* Interface Failure between Base Skin and Flange

Figure 20 Y-Frame Designs Demonstrated Significant Increase in Pull-Off Strength (RTD Tests)

| Frame Configuration | Initial Failure |                 |                               | Final Failure |                 |                    |
|---------------------|-----------------|-----------------|-------------------------------|---------------|-----------------|--------------------|
|                     | Load (lbs/in)   | Deflection (in) | Failure Mode                  | Load (lbs/in) | Deflection (in) | Failure Mode       |
| Blade               | 372             | 0.155           | Comp. Failure in Lower Skin   | 417           | 0.255           | Interface Failure* |
| 45° Y-Frame         | 530             | 0.145           | Skin Surface Failure at Clamp | 746           | 0.270           | Interface Failure  |
| 60° Y-Frame         | —               | —               | —                             | 615           | 0.192           | Interface Failure  |

\* Interface Failure between Base Skin and Flange

Figure 21 Y-Frame Designs Demonstrated Significant Increase in Pull-Off Strength (ETW Tests)

RTD Blades - Each of the RTD blade specimens exhibited a distinctive initial failure which was associated with a radius crack within the angle ply pack. This initial failure was visually observable and correlated exactly with initial load drop-off. Propagation of the initial crack continued through the remainder of the loading sequence. A second failure was observed to correspond to the loss of the bond between the skin and frame laminates as the radius crack progressed to the interface. The final portion of the load/displacement curve indicated the continued loss of the interface as a result of the crack propagation with the second failure load of lower magnitude than the crack initiation load. Cracks associated with initial and final failure are readily observable for a typical failed specimen shown in Figure 22.

RTD 60° Y- Frames - Initial failures in the Y-frames were associated with compression failure of the skin laminates at the clamped boundary condition. This failure is the cause of the decreased section stiffness and resulted without any indication of interface failures. Final failures typically occurred after a slight increase in load and substantially larger deflections.

Increases in the final failure load for a Y-frame element over the blade configuration are a result of decreased interlaminar tension stresses through redistribution of the pull-off load into combined interlaminar tension (ILT)

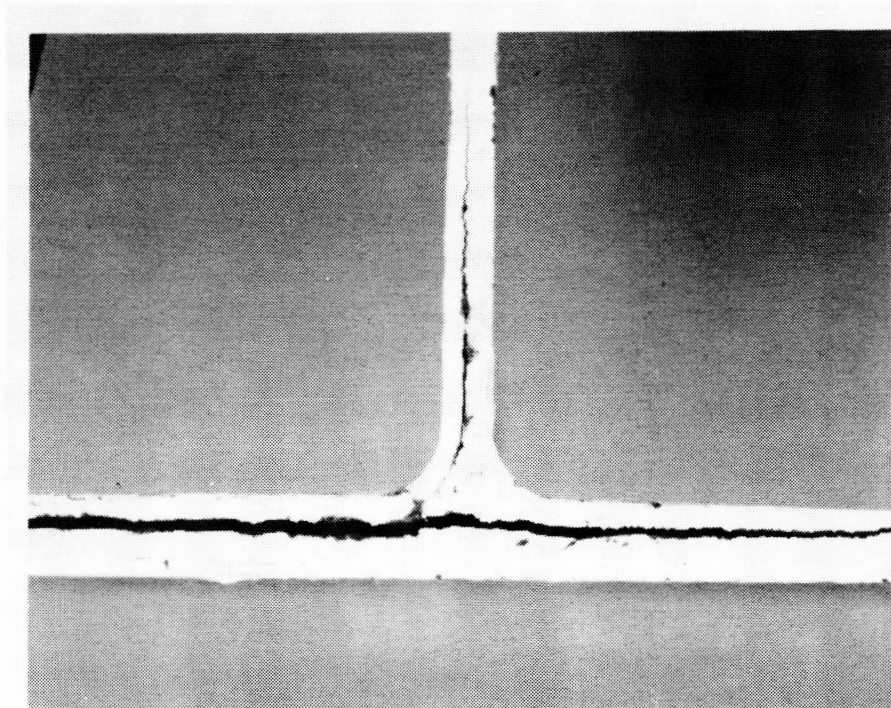


Figure 22 RTD Blade Elements Experienced Skin/Flange Interface Separation

and interlaminar sheal (ILS). In addition, ILT stress within the upstanding flange laminates is reduced due to the increased radius associated with this geometry. Observation of the final failure for the 60° element, Figure 23, showed crack initiation occurred as a result of interlaminar tension within the IML skin and resulted in catastrophic failure of the section.

RTD 45° Y-Frames - Nearly identical specimen response as observed for the 60° element was observed for the 45° geometry. This similarity includes initial non-linearity, minor variations due to slip, initial failure due to skin compression and associated reduction in modulus, and substantially higher ultimate pull-off strength. However, the 45° elements all showed continued load carrying capacity following the secondary ILT failure. This additional loading is more pronounced than any observed in the 60° elements and is associated with geometry and load redistribution effects. The larger open angle of the IML skin and further increase in corner radius decreases the ILT stresses within the IML laminate. In addition, loading continues to be redistributed into predominantly ILS stresses, as in the 60° frame. Failure observation for the 45° element showed crack initiation occurred as a result of ILT within the IML skin as previously witnessed in the 60° specimens. However, final failure eventually resulted from propagation of this crack to the interface and continuation of this delamination primarily through a shear mechanism. This crack growth proceeded at a slower rate than observed for either of the previous designs.

ETW Blades - Failure of the blade specimens became a more complicated phenomena for the elevated temperature wet condition. A higher degree of non-linearity in the initial load/displacement curve was recorded. Initial failures occurred as the result of compression in the lower skin. Both of these results are attributed to the softening of the material associated with the ETW condition. Very shortly after initial failure, a secondary failure in the interface was observed. Crack initiation occurred at the identical location as for the RT tests. Following crack initiation, secondary failure load carrying capacity of the frame attachment continued beyond the initial failure loads. Crack propagation was observed to progress far more slowly due to the plastitization of the matrix material. Eventual loss of all strength was associated with the loss of a significant portion of the flange to skin interface (approximately 50%).

Failure loads for all of the blade specimens considerably higher than the RTD tests. Similar results for lap shear specimens have been reported in previous work conducted at MCAIR. The mechanism for this increase in ETW strength is identical for both test situations. Elevated temperature response for the interface material shows a decrease in both stiffness and strength. However, an associated increase in ultimate strain is associated with the material due to its plastic response. The lap shear data has shown that the associated energy that the material is capable of absorbing prior to failure is significantly increased due to the large plastic deformations that become possible for the ETW material. The effect of the plasticity is to lower the peak stresses at the ends of interfaces while increasing the stresses across the remainder of the interface. This results in an increase in the total load



carrying capacity of the interface. The plasticity also accounts for the non-linearity observed during the initial portion of the load/displacement plot.

ETW 60° Y-Frames - The 60° frame elements exhibited the same degree of non-linearity that was observed for the blade specimens. Skin compression failures occurred at the clamp location but to a much lesser extent than previously encountered for RT testing. No indication of skin compression failures beneath the frame attachment point was evident during the test or in the load/deflection history. A typical failed specimen is shown in Figure 24. Failure at the IML to OML interface occurred with immediate propagation of this failure to the clamp locations. No failures were witnessed within the IML skin due to interlaminar tension as observed in the RTD testing.

The increase in ultimate loading over the baseline blade configuration was also observed for these specimens and again indicates the advantage of redistributing load into ILS and ILT through the Y-frame design.

ETW 45° Y-Frames - Failure of the 45° frame specimens was more severe than had been witnessed in the 60° frame specimens and is attributed to the increased deflection observed in these specimens. Compression skin failure occurred at the clamped boundary. Final failure again occurred with the total failure of the interface due to primarily shear loading. The full failure of the interface is another indication that the plasticity of the matrix material is allowing the entire interface to carry more load than in the room temperature cases. This increased plastic loading prohibits the interface from resisting any cracks which initiate.

Overview of Frame Element Tests - Testing of the fastenerless frame attachments provided the necessary experience with these designs to allow for risk reduction in future scale up to subcomponent and full-scale articles. Of primary importance has been the demonstration of significant increases in the pull-off strength of frame attachments that can be achieved through Y-section designs. Through the examination of two geometries an initial understanding of the effects of geometry on load redistribution and strength improvements has been gained. Gains on the order of 50% were achieved for designs involving a 60° frame element as compared to the baseline blade. An additional improvement of 10% was obtained when the interior angle of the Y-section was reduced to 45°. These improvements are based upon initial failure loads for all of these sections and are summarized in Figure 25. Increases in section deflections were also evident in the test data. This fact points out the importance of including a local pad-up beneath the Y-section for future design developments.

Elevated temperature testing demonstrated the improvements in performance associated with a thermoplastic material subjected to interlaminar shear and tension. These increases are justified when considering the potential for plastic deformation of the matrix material in the interface. Some concern over this type of response will undoubtedly remain until an understanding of the possible fatigue response is investigated.

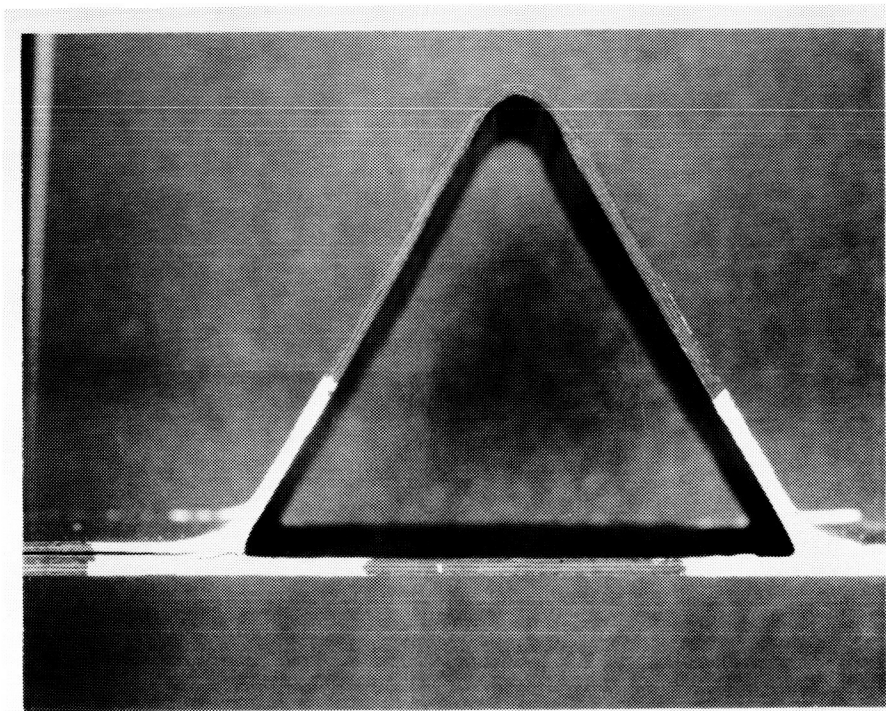


Figure 23 RTD 60° Y-Frame Elements Experienced Skin/Flange Interface Separation and Interlaminar Tension Failures

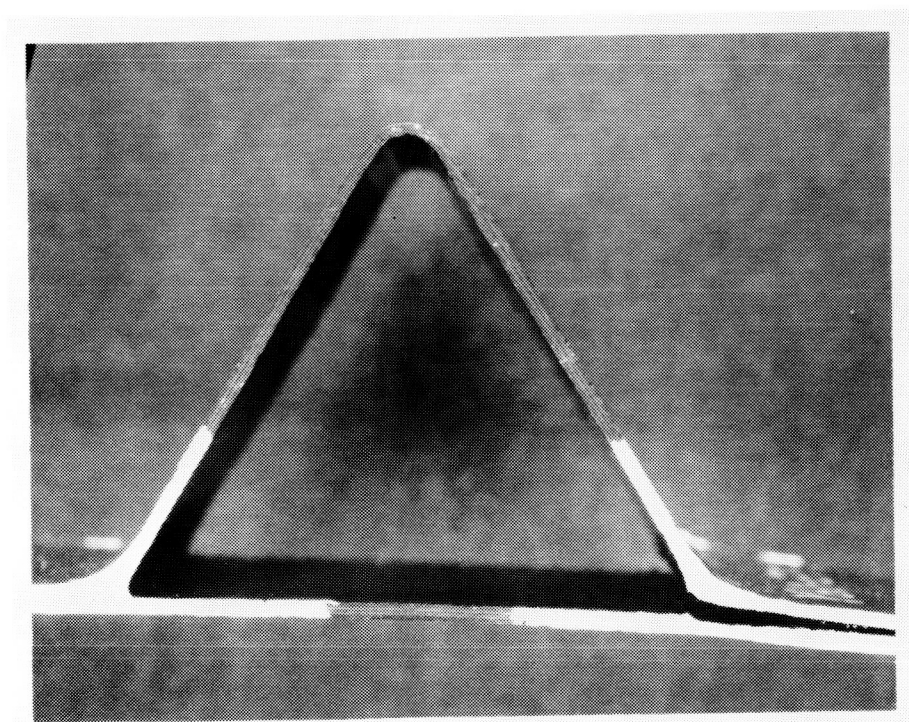


Figure 24 ETW 60° Y-Frame Elements Experienced Skin/Flange Interface Separation



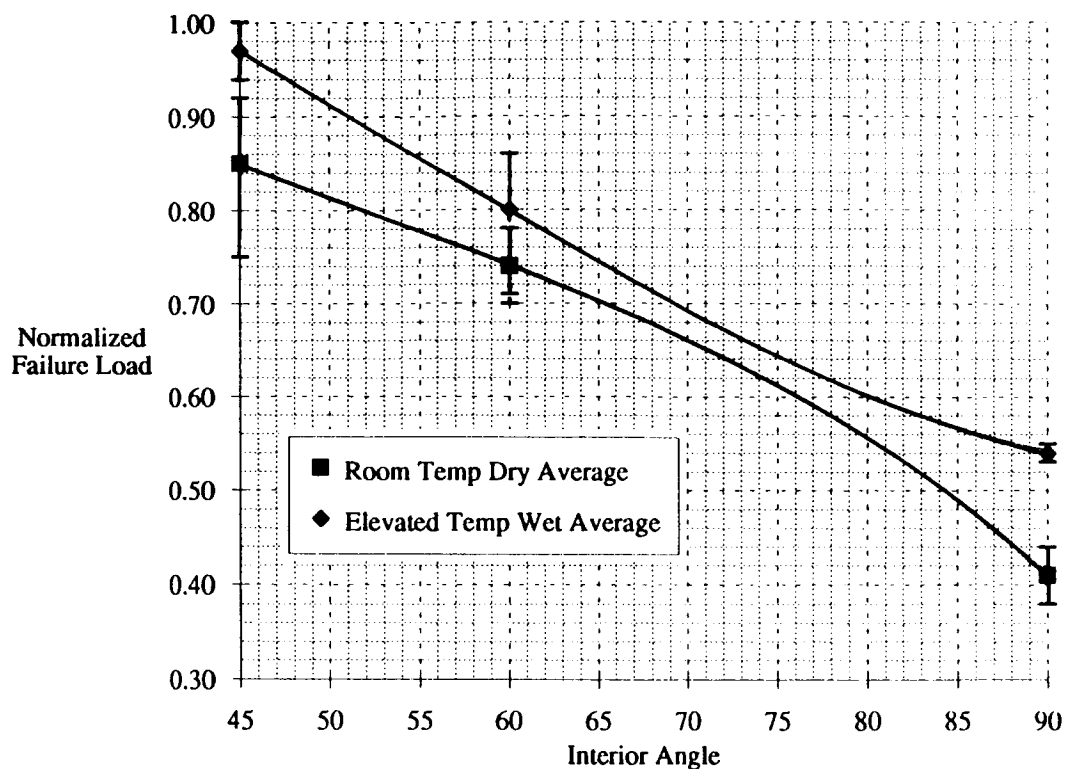


Figure 25 Performance Improvements Associated with Y-Frame Designs

With regard to testing, it is evident that future attempts must identify a means of load introduction which will not produce an interference between the load mechanism and the lower skin section. It is also recommended that skin thicknesses be increased to prohibit bending failures at the grip locations. These changes to the base laminate should include the addition of the previously mentioned pad-up region beneath the Y-section. A means of eliminating slipping beneath the clamping device should be found. As an alternative, the use of simple (or roller) supports might be examined. This problem might also be advantageously influenced by the increased skin thickness already proposed. Finally, it should be mentioned that the use of videotaping in conjunction with the use of white-out on the specimen edges proved to be invaluable in correlating test results with observed failure modes.

## Lug Elements

Lug Element Designs - In aircraft design, single-pinned joints (i.e., lugs) play a key role in the transmission of large loads between major structural components. Thus, the potential for substantial weight reductions exists in the use of composite materials for these highly loaded structures.

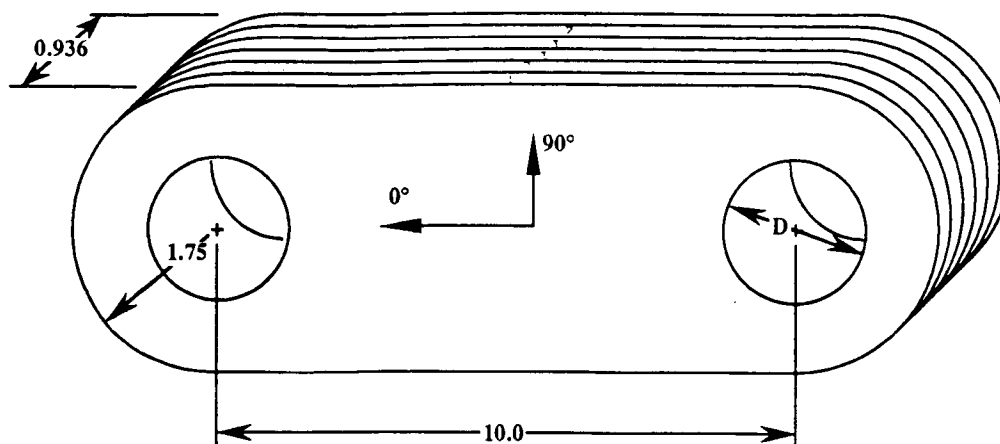
Unlike most element design efforts, the design of lug elements was not focused on identifying an optimal design for application to the fuselage structure and loading being considered. Instead, the designs were chosen to provide a fairly comprehensive set of test cases to which analytical developments could be compared. This form of building block approach was adopted due to the complexities involved in the design of both lugs and thick composite sections. The end result would be an analysis package with the means of rapidly comparing multiple design strategies later as the definition of the full-scale article became solidified.

The design of thermoplastic lug test articles focused on providing specimens which could validate the most critical portions of the analysis. Pin bending effects and a study of the different failure modes were targeted by the D/MI team as the primary areas of concern. A means of separating these design variables and maintaining constancy for all other design parameters became the focus of the team.

Previous in-house efforts determined that ply stacking sequences can affect the pin bending response of a lug. By varying the through-the-thickness stiffness distribution of the laminate, peak stresses could be reduced, thereby minimizing pin bending effects. In order to explore this potential for increased performance and to validate the methodology's ability to predict it, three lug layups were chosen which resulted in different pin bending responses while maintaining nearly identical in-plane response. This was accomplished by varying the stacking sequence of the six sublaminae used in the manufacture of each lug. Maintaining the in-plane response was necessary to allow for an investigation of the other design parameter, failure mode.

Failure mode dependence on lug geometry is well established for metallic designs and was anticipated to be the case for the composite lugs also. Lug geometries were chosen to provide two distinct modes of failure based on a metallic analysis using smeared in-plane properties for the composite lugs. External geometries on all the lugs were held constant ( $W = 3.5$ "), while the hole diameter was varied to 1.0" or 1.75". This resulted in different  $W/D$  ratios (3.5 and 2.0 respectively), and was anticipated to provide bearing failures in 50% of the specimens, while the remainder would experience net-section failures. A summary of the lug designs used for analysis correlation is shown in Figure 26 and represents the approach for exploration of critical design issues in composite lug analysis.

The analysis development pursued under this program was originally reported in Reference 2 and reported in more detail in a subject paper at this conference.



Note : All dimensions are inches.

| Lug Specimens | Quantity | Hole Diameter (in) | Sublamine Stacking Distribution* |             |             | Effective Layup |
|---------------|----------|--------------------|----------------------------------|-------------|-------------|-----------------|
|               |          |                    | Sublamine 1                      | Sublamine 2 | Sublamine 3 |                 |
| Static 1      | 4        | 1.00               | (47/40/13)                       | (47/40/13)  | (47/40/13)  | (47/40/13)      |
| Static 2      | 3        | 1.00               | (47/40/13)                       | (34/53/13)  | (60/27/13)  | (47/40/13)      |
| Static 3      | 4        | 1.00               | (60/27/13)                       | (47/40/13)  | (20/67/13)  | (42/45/13)      |
| Static 4      | 4        | 1.75               | (47/40/13)                       | (47/40/13)  | (47/40/13)  | (47/40/13)      |
| Static 5      | 4        | 1.75               | (47/40/13)                       | (34/53/13)  | (60/27/13)  | (47/40/13)      |
| Static 6      | 4        | 1.75               | (60/27/13)                       | (47/40/13)  | (20/67/13)  | (42/45/13)      |

\* Sublamine Stacking Sequences are as follows:

(47/40/13) :  $[0_2/45_2/90_2/-45_2/0_2/45/0/-45/0]_s$

(34/53/13) :  $[45_2/0_2/-45_2/90_2/45_2/0_2/-45_2/0]_s$

(60/27/13) :  $[0_2/45/0_2/-45/0_2/45/0_2/-45/90_2/0]_s$

(20/67/13) :  $[45_2/0/-45_2/90/45_2/0/-45_2/90/45/0/-45]_s$

Figure 26 Configuration and Stacking Sequence of Lug Elements Used to Verify Analytical Developments

Lug Element Fabrication - The lugs were fabricated using AS4/APC-2 unidirectional tape since it was readily available early in the program and the lugs were primarily being tested for analytical model verification. Tooling for the lug specimens consisted of simple project plates with steel plates positioned to allow for expansion during consolidation.

The lugs were fabricated from eighteen 30" x 16" sublaminate panels of four different 30 ply lay-ups. The sublaminates were consolidated in a hydraulic press. Six sublaminates were stacked to form the three different 180 ply stacking sequences which were coconsolidated in the autoclave. Excellent consolidation of the sublaminates was verified by photomicrograph inspection. Panels with the final lug lay-ups were C-scanned to ensure their quality.

Water jet cutting was used to obtain four lugs from each of the panels, Figure 27. The finish associated with this cutting procedure was acceptable as the final external finish on the test articles. Initial lugs showed a minor defect on the external surface at the beginning of the radius. This was associated with the initiation of the WJC process and to avoid this stress riser, the cutting pattern was altered to begin in the middle of the lug rather than at a radius.

The pin holes were initially WJC and secondarily milled to ensure the necessary tolerances of  $\pm 0.005$  inches. C-scans were taken of the finished lug to ensure the cutting procedure had not induced any delamination. The final lug specimens, as typified in Figure 28, were found to be void-free.

Lug Element Tests - Room temperature testing was performed with a load rate of 500 lbs/sec, and all lugs were tested in as received moisture condition. The testing setup is shown in Figure 29. In order to keep the gap between the lug and loading clevis surface at 0.1", which was used in the analytical models, special bushings were fabricated. A different gap size might cause unexpected failure loads due to different pin bending effects.

The lugs with 1.75" diameter holes ( $W/D = 2.0$ ) exhibited a catastrophic fiber failure at the net section, whereas the lugs with 1.0" diameter holes ( $W/D = 3.5$ ) showed permanent yielding around the hole prior to shear/bearing failure. The initial bearing failure load was determined by observing the behavior of axial strain data from rosettes located at 0.5" away from the edge of the 1.0" hole and 0.25" away from the 1.75" hole at the center line of specimens. Load versus strain data indicates that axial strain decreases with associated material failure ahead of the pin for the 1.0" hole lugs, Figure 30. Typical failed specimens are shown in Figure 31. These results for the six lug configurations are summarized in Figure 32 along with analytical predictions.



Figure 27 Thick Composite Lug Elements were Efficiently Machined Using Abrasive Waterjet Cutting

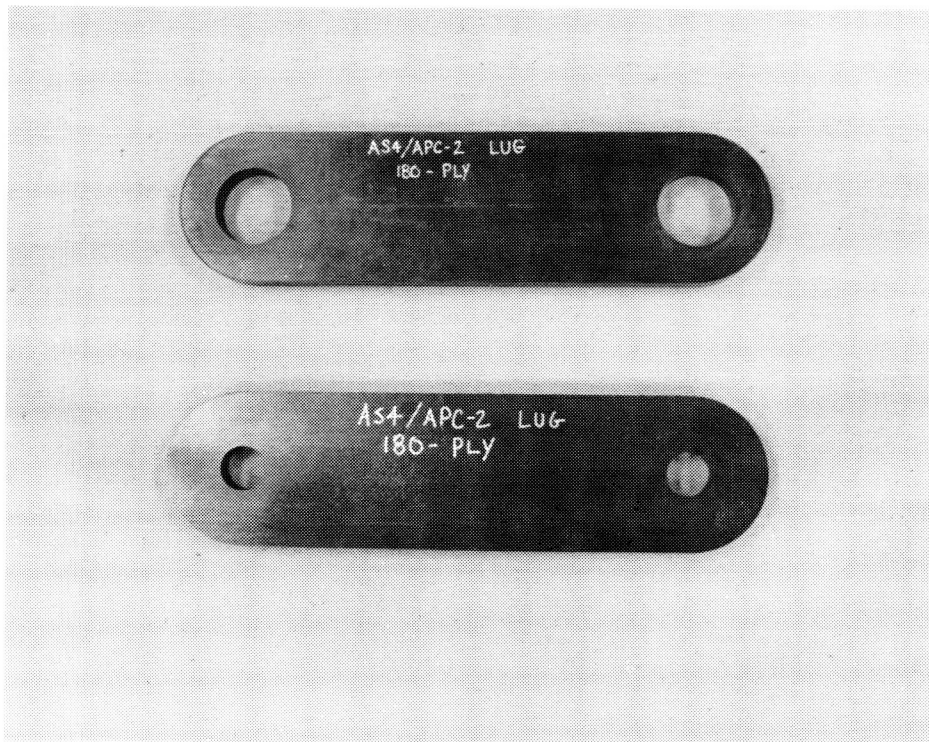


Figure 28 Two W/D Geometries were Used to Demonstrate Distinctly Different Failure Modes

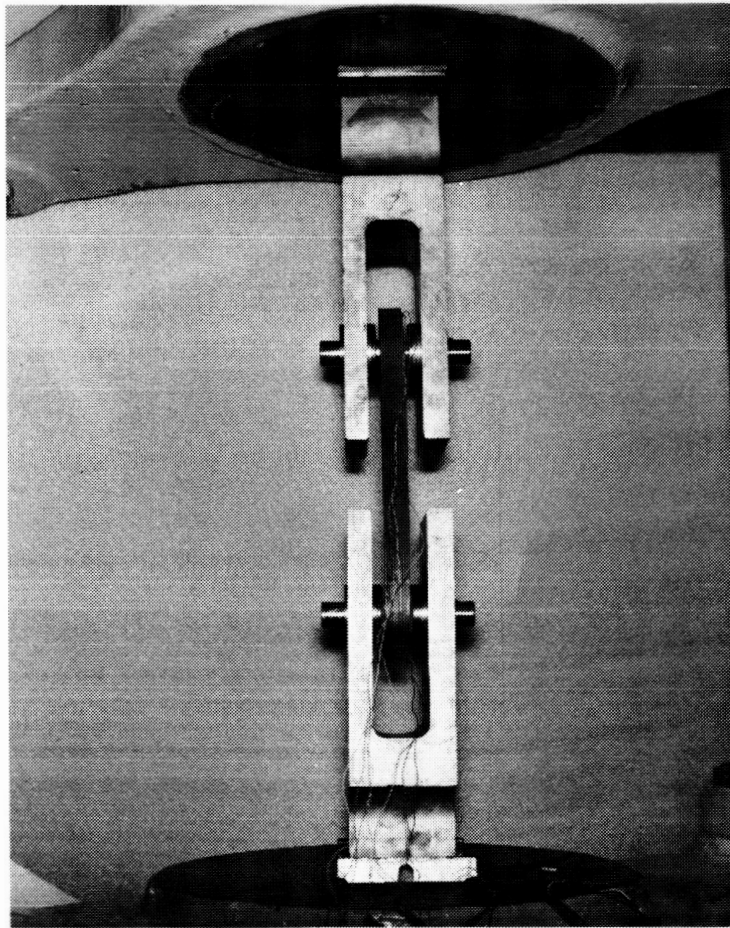


Figure 29 Lug Element Test Set-Up Simulated In-Plane Loading

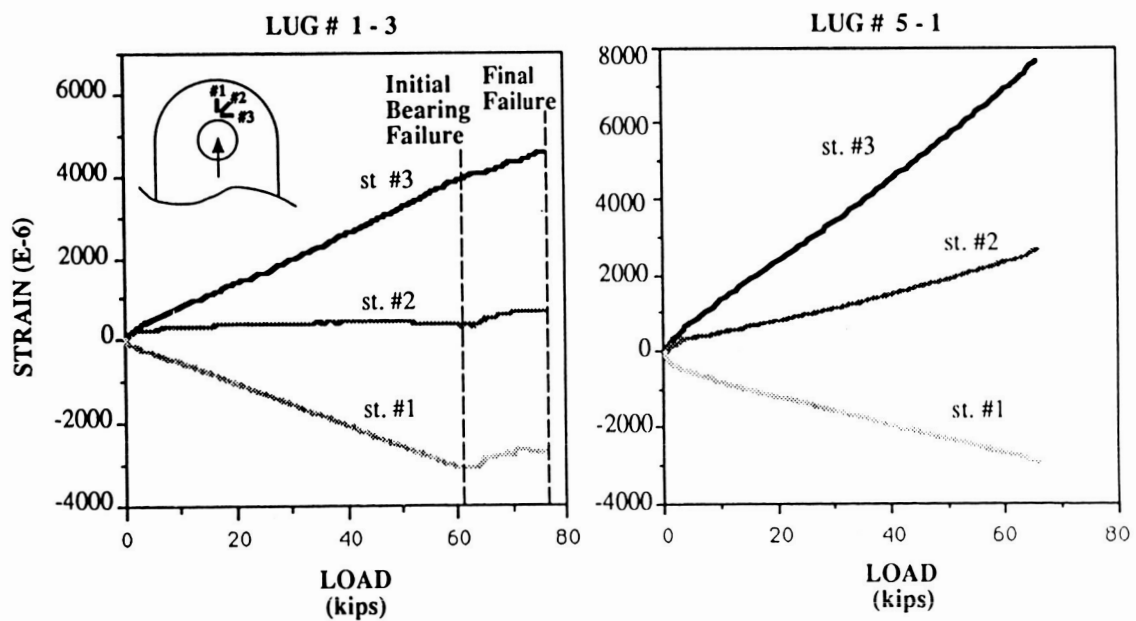
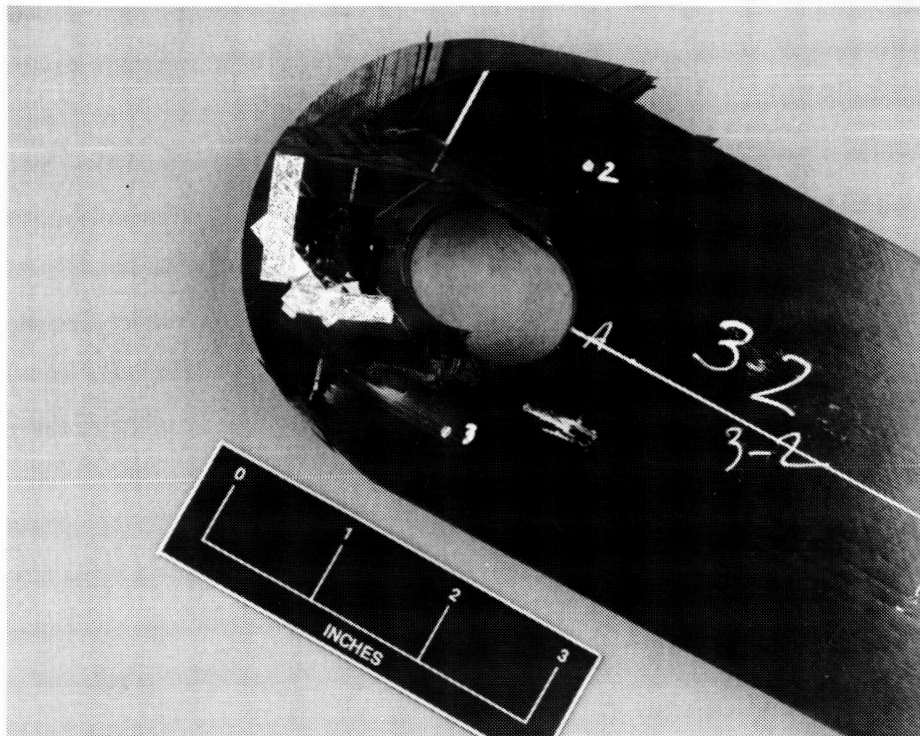
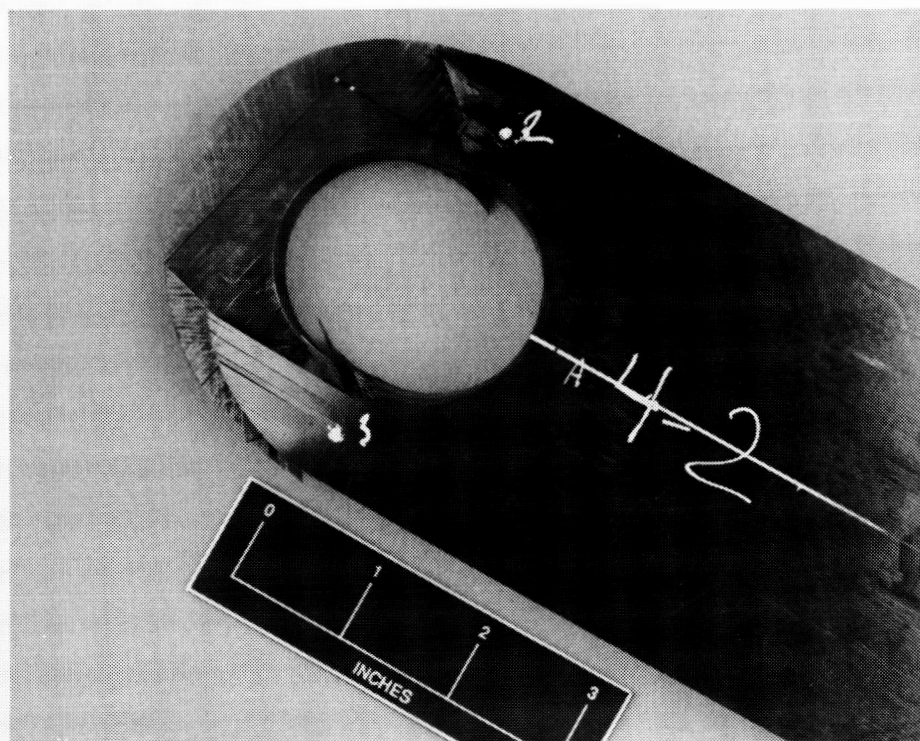


Figure 30 Lug Element Load vs Strain Data Provided Accurate Determination Of Failure





Bearing and Shear-Out Failure Observed for 1.0" Diameter Lugs



Net-Section Failure Observed for 1.75" Diameter Lugs

Figure 31 A Range of Lug Element Failure Modes were Demonstrated

| Lug Specimen | Quantity Each | Hole Dia. (in) | Pred. Failure Load (kips) | Average Test Results (kips) | Standard Deviation |
|--------------|---------------|----------------|---------------------------|-----------------------------|--------------------|
| # 1          | 4             | 1.0            | 64.2                      | 60.1 (76.7)*                | 4.3 (2.2)          |
| # 2          | 3             | 1.0            | 65.9                      | 62.3 (74.3)*                | 6.9 (3.0)          |
| # 3          | 4             | 1.0            | 57.3                      | 62.2 (74.7)*                | 7.0 (2.0)          |
| # 4          | 4             | 1.75           | 66.9                      | 69.3**                      | 3.1                |
| # 5          | 4             | 1.75           | 66.5                      | 68.7**                      | 2.9                |
| # 6          | 4             | 1.75           | 61.5                      | 66.8**                      | 1.8                |

\* Initial Bearing Failure Load (Final Failure Load)

\*\* Failure Load

Figure 32 Close Agreement Between Predicted and Actual Lug Element Strengths was Achieved

#### Rolled Formed Stiffeners

Vanguard Composites Company (Anaheim, CA) was subcontracted to fabricate 45 feet of roll formed hat stiffeners. These stiffeners were to be originally used in subcomponent fabrication using the fiber placement process, but served as a manufacturing demonstration due to the redirection.

The thermoplastic hat stiffener consisted of 7 plies of IM7/ITX which incorporates a 4 ply drop-off in the flanges representative of the fiberplaced panel design. A single layer of 3 mil neat PEEK film is incorporated on the IML of the flanges for future bonding trails. C-scans of the first article hat stiffener showed a lack of consolidation in the flange ply drop-off area as well as in the stiffener walls in the area of the upper radius. The first article also had visible roller lines transverse to the stiffener's length and dry patches on the IML flanges where the neat resin film had thinned out.

A second article showed slightly better consolidation in the ply drop-off areas but still had poor consolidation in the stiffener walls. The roller marks were eliminated and the neat resin film application looked much better on the second article. It was discovered by Vanguard that the roller for consolidation of the stiffener walls had a 1/2° mismatch which was most likely the reason for the inadequate consolidation in this area. The problem was corrected and fabrication continued.



Forty-five feet of roll formed hat stiffeners was received from Vanguard Composites Company. A typical section is shown in Figure 33. NDT results from a random sample of the hat stiffeners show a lack of adequate consolidation in the flange area. The constant thickness portion of the flanges showed significantly better consolidation. Based on these observations, it is felt that roll forming of constant cross-section, thermoplastic composite stiffeners with uniform flanges is a viable manufacturing process. However, tapered flanged stiffener concepts would require additional development.

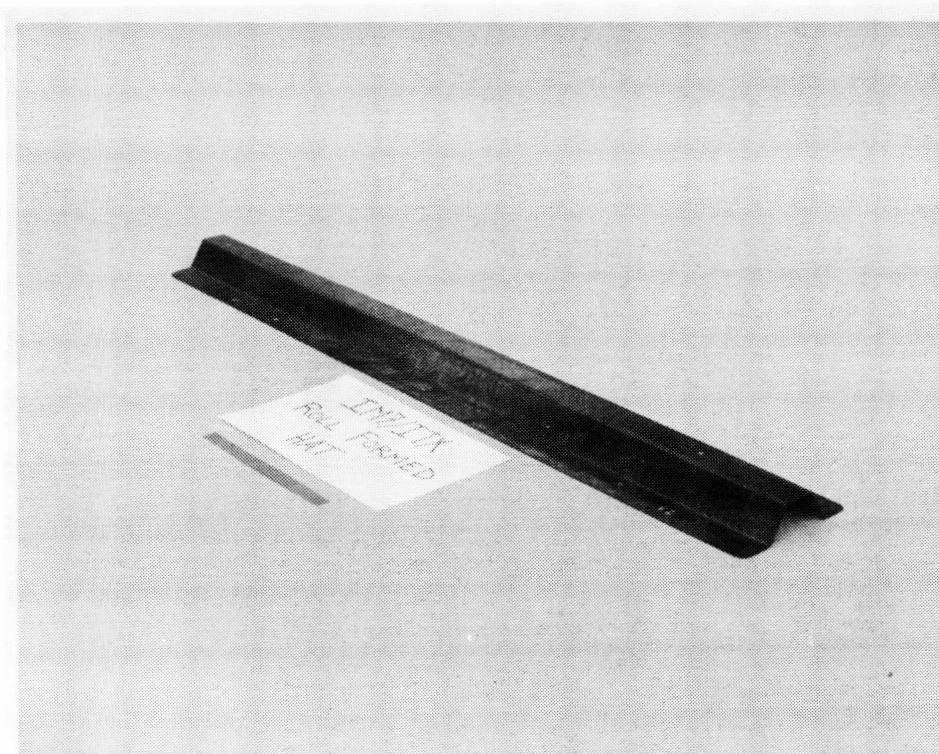


Figure 33 Rolled-Formed Hat-Section Stiffeners Demonstrated Potential for Low Cost Manufacturing

## CONCLUSIONS AND RECOMMENDATIONS

Thermoplastic composite development pursued in this program, while directed toward fighter aircraft structure, is equally applicable to commercial vehicle structure. These developments focused on critical composite issues associated with primary fuselage structure, fastenerless moldline, upper fuel cell cover structure, and thick lugs representative of those on carry-through bulkheads. Activities by a D/MI team were carried out in the areas of structural mechanics, manufacturing concepts development, and structural validation. A summary of element design, fabrication, and structural testing was presented in this report.

Elemental manufacturing verification trials produced valuable lessons learned. In the SDCC Y-frame activities simple tooling modifications such as blended stiffener mandrel and pressure box ramp intersections coupled with aluminum diaphragms eliminated diaphragm ruptures. Polymeric diaphragms are still desirable from a cost point of view, but as yet do not have the necessary elongation properties needed for complex forming.

A one-step SDCC process was a program goal, however, ply dragging/wrinkling problems necessitated going to a two-step process which yielded production quality parts. While the two-step process provides cost savings over conventional three-step diaphragm forming, the one-step technique should still receive industry attention since additional cost savings could be realized.

The development of a positive drive tooling concept for the blade frame elements came about as a need to correct inadequate intersection consolidation pressure inherent in the original tooling. The addition of selected keyways to the tooling allowed for segmented tooling details to be directed (positively driven) during pressure application resulting in a first-time quality part. In addition, a previously rejected blade element was reconsolidated to production quality in the modified tooling demonstrating the potential cost savings associated with the ability to reconsolidate thermoplastic composites.

Roll-forming may be a viable technique for producing long, relatively constant cross-section stiffeners. While sections produced in this effort did not reach production quality consolidation, overall cross-section geometrical tolerances, straightness in length, and repeatability were quite good. Investigations in this program revealed that while there is still some development necessary, this approach should receive additional industry attention.

Thick laminates (1.0 inch) were manufactured to production quality by consolidating a series of sublaminate. An additional benefit of this effort was the demonstration of abrasive water jet cutting as an effective means of machining thick composites. Edge surface finish was found to be very acceptable for high tolerance areas requiring only modest surface reaming or finishing for low tolerance areas.

Significant pull-off strength increases were demonstrated in the Y-frame concepts compared to conventional blade design. Testing of the fastenerless frame concepts has provided the necessary experience with these designs to allow for risk reduction in future scale-up to subcomponent and full-scale structure.

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COMPOSITE INTERMEDIATE CASE  
MANUFACTURING SCALE-UP FOR ADVANCED ENGINES

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SUMMARY

This Manufacturing Technology for Propulsion Program developed a process to produce a composite intermediate case for advanced gas turbine engines. The method selected to manufacture this large, complex part uses hard tooling for surfaces in the airflow path and trapped rubber to force the composite against the mold. Subelements were manufactured and tested to verify the selected design, tools and processes. The most significant subelement produced was a half-scale version of a composite intermediate case. The half-scale subelement maintained the geometry and key dimensions of the full-scale case, allowing relevant process development and structural verification testing to be performed on the subelement before manufacturing the first full-scale case.

INTRODUCTION

The objective of this Air Force Manufacturing Technology for Propulsion program is to establish manufacturing methods for the fabrication of a composite intermediate case for the next generation of fighter aircraft engines. The work is being performed by Pratt & Whitney and DuPont under the direction of Mr. Kenneth Ronald of the Air Force Manufacturing Technology Directorate, WL/MTPN, under Phase Va of Contract F33615-85-C-5152.

The intermediate case in today's military gas turbine engines is a large complex titanium casting that is required to support front engine bearings, maintain blade clearances, handle multiple engine loads, and direct airflows into both the high-pressure compressor and the bypass duct. Figure 1 shows the location of the intermediate case in a typical jet engine. Figures 2 and 3 show the front and aft sides of a typical titanium intermediate case. The complexity of the part and the requirement that the part carry a substantial structural load presents a significant challenge to design and manufacturing.

The intermediate case also presents a materials challenge. Airfoils located further from the engine inlet, beyond the second stator vane, are subject to prolonged exposure to air at temperatures up to 371 C (700F) and at pressures of several atmospheres. Air oxidation tests have been run at P&W at a temperature of 371C (700F) and a pressure of 4 atmospheres, shown in Figure 4. These oxidation tests have shown that Avimid N, a fluorinated polyimide manufactured by DuPont, has a definite advantage over PMR-15 in atmospheric oxidation resistance.

#### TECHNICAL APPROACH

The approach used by DuPont, the supplier of Avimid N and the subcontractor to Pratt & Whitney in the intermediate case program, was to develop tooling and procedures for subelements and then progress to the manufacture of a full-scale case.

The process evaluation task was divided into four major subtasks:

- Definition of subelements

- Development of process parameters for fabrication of subelements

- Fabrication of test subelements

## Testing of subelements

The subelements selected were (1) half-scale rings, (2) struts and (3) a half-scale case with a half-scale outer ring and three full-size struts.

Half-scale Ring. The fabrication approach uses hard tooling for surfaces in the engine airflow path and a high-expansion rubber to generate and transmit the compaction pressure in all areas not in the path of the airflow. A high-temperature silicone rubber was chosen as the high-expansion material to generate pressure for the ring and other subelements.

Eight half-scale diameter outer ring subelements, measuring 45.7 cm (18 in.) in diameter, were manufactured to validate the fabrication approach. The first Avimid N ring, shown in Figure 5 exhibited good resin flow control and fair material consolidation. The results demonstrated that this fabrication approach is viable, and with some modifications to the approach, well-consolidated subelements could be produced. The expansion behavior of the rubber and prepreg layup techniques were investigated during the manufacture of the next seven rings.

Strut Subelement. The first strut subelement was laid up using preliminary ply shapes and cured to demonstrate the suitability of using rubber inserts in the tools to generate the consolidation pressure. Rubber expansion calculations had indicated that the relatively thick leading and trailing edges of the strut could pose consolidation difficulties. A simple strut subelement tool was designed to explore and resolve these problems. The tool and resulting strut are shown in Figure 6. Tool closing difficulties, associated with the bulk factor of prepreg materials, indicated that modifications of the ply shapes and layup techniques would be required.

A number of struts were laid up and cured with various cure cycles. All of the strut elements had poor consolidation compared to the ring described in the preceding section because the pressure developed by the rubber apparently varied from cycle to cycle. A separate study of the relationship between pressure increase of the rubber with temperature increase was linear, but the slopes of the lines were 30 degrees, 45 degrees, and 60 degrees.

A high-temperature pressure transducer was procured to perform direct measurement of cure pressure generated by the rubber expansion material. A consolidation pressure of 12.42 MPa (1800 psi) was measured. Devolatilization times were also adjusted to achieve adequate resin flow for good consolidation. These process modifications resulted in the well-consolidated strut shown in Figure 7.

Half-scale Subelement. The next major step in the process development phase was to design and manufacture a half-scale version of the full-scale case.

The half-scale subelement has a diameter half the size of the full-scale intermediate case and contains three struts instead of six (Figures 8 and 9). The struts in the subelement are approximately full-size axially, radially and in width. This geometry maintains key dimensions of the full-scale part such as the strut chord and the curved distance between struts. Features of the subelement are:

An outer shell produced from three arced segments with composite flanges and boxed reinforcements at each strut.

Three hollow airfoil-shaped struts with splitter tabs attached at the midspan, aft edge. The struts were

approximately full-scale in both chord and radial span.

A conical inner shell with flanges for the bearing supports and boxed reinforcements at each strut.

The subscale case was thermally cured in a hard mold shown in Figure 10 that controlled the aerodynamic surface features within the gas flowpath areas. During the cure, the mold was loaded axially in a press. A silicone rubber compound was used inside the hollow struts and around the inside diameter and outside diameter surfaces to pressurize the flowpath areas against the mold during the cure process. The flange features were then machined into the composite material.

The test article detailed in this report had a circumferential linear manufacturing defect through the midspan of the outer shell, between strut No.'s 1 and 3. Another manufacturing problem that occurred was intrusion of the silicone rubber pressurizing material into other strut areas during the cure process. This caused delaminations within the struts and weakened the splitter tab attachments. Although, the extent of rubber intrusion was unknown, the case was considered adequate for structural testing. However, observations during and after the tests revealed that the rubber intrusion was much more extensive than expected. This problem was addressed during the manufacture of the full-scale case by adding sacrificial protective plies .

Since the half-scale subelement was a scaled down version of the full-scale case, it was tested extensively to verify the design and structural strength predictions for the full-scale case. The test loads used for the subelement were based on predicted engine operation and flight maneuver conditions. The test rig, shown in Figure 11, is capable of applying loads of 90,718 kg (200,000 lb). The load conditions tested were:



|                               |          |
|-------------------------------|----------|
| Combined Bearing Radial Loads | Limit    |
| Outer Flange Axial Load       | Limit    |
| Outer Flange Combined Loads   | Limit    |
| Splitter Axial Thrust Load    | Limit    |
| No. 2 Bearing Radial Load     | Limit    |
| Splitter Axial Thrust Load    | Limit    |
| No. 2 Bearing Radial Load     | Ultimate |
| No. 3 Bearing Radial Load     | Ultimate |
| Splitter Axial Drag Load      | Limit    |
| Combined Bearing Radial Loads | Ultimate |

Limit load requirements were successfully met by the part. Ultimate load test performance was less than anticipated but the problems were attributable to rubber intrusion, not design or material inadequacy.

Full-scale Case Manufacture. After completing the subelement testing, full-scale cases were manufactured. Over 3000 individual plies were cut on a Gerber cutter and laid up on subelement preform tools. The subelements were then assembled into the full-scale case preform tool and the final plies were added to complete the case.

The first full-scale case manufactured, a tool proof case, had well consolidated struts and inner ring but flanges and the outer ring had areas that were poorly consolidated. Cure temperature ramp rates were adjusted and thermal blankets were added to the outer ring tool to correct this problem.

The second full-scale case was well consolidated but the splitter tabs were damaged during fabrication. This problem was corrected by modifying the cure tools to aid tool disassembly and ensure complete strut tool closure during cure.

The third full-scale case has been manufactured and tested. The test results will be reported in the final report of Air Force contract F33615-85-C-5152.

#### CONCLUSION

This program has demonstrated that a complex part such as an intermediate case for gas turbine engines can be produced from a high temperature composite material. The half-scale subelement, which maintained key dimensions and contained the geometrical complexity of the full-scale part was a significant factor in developing this manufacturing technology. Tests performed on a half-scale subelement verified that a composite case could meet the rigorous structural requirements of this important engine component.

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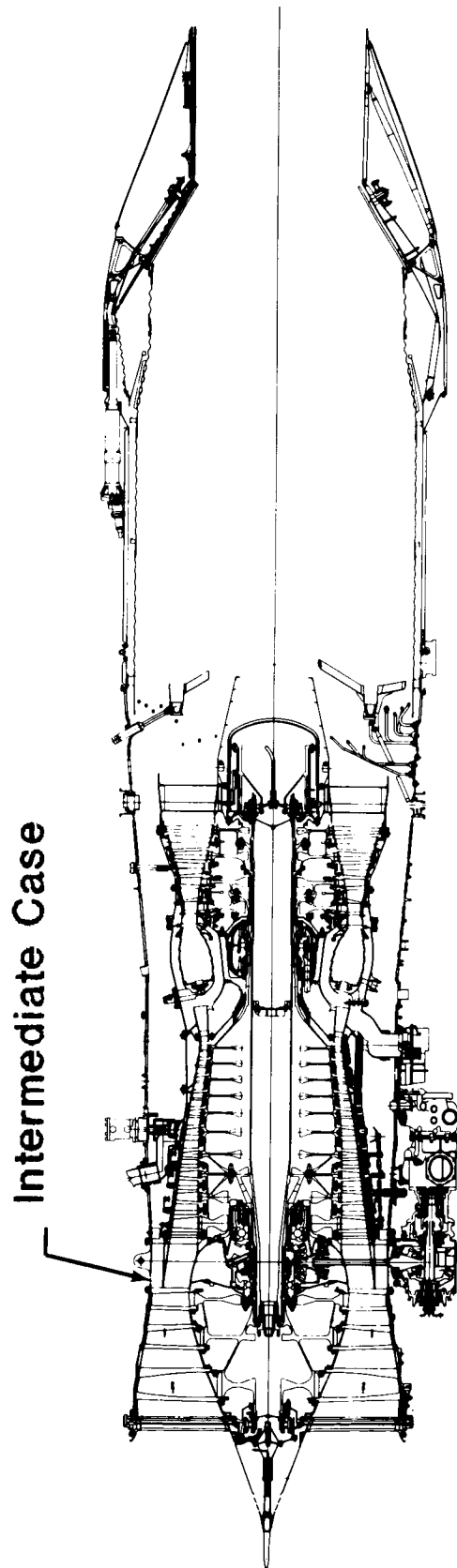


Figure 1. Engine Cross Section

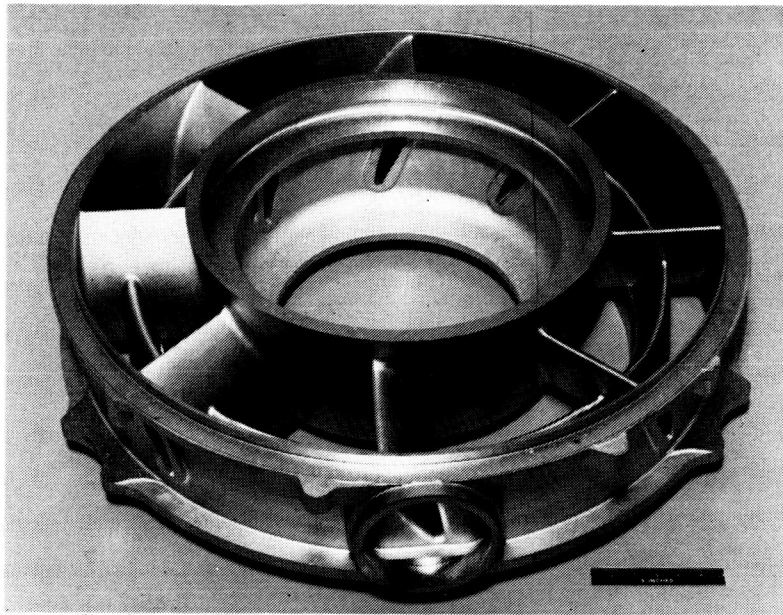


Figure 2. Front Side of Typical Titanium Intermediate Case

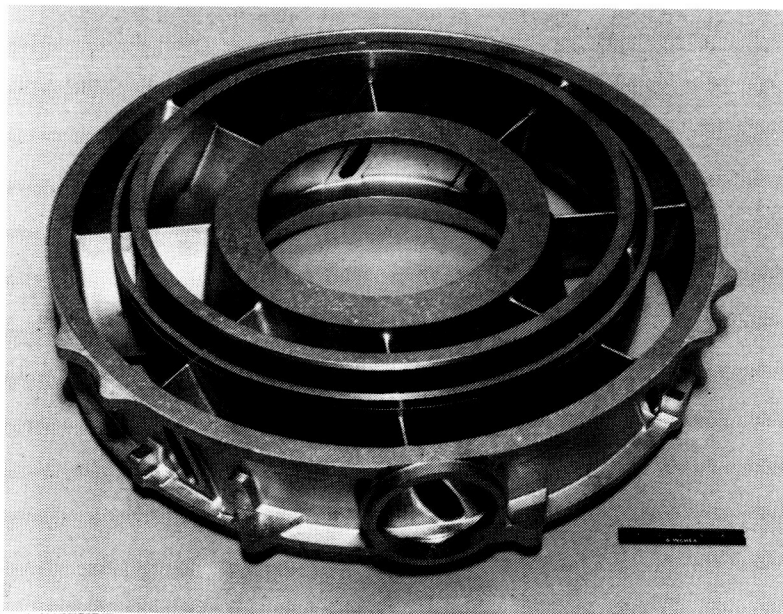


Figure 3. Aft Side of Typical Titanium Intermediate Case

## Oxidation Weight Loss Of Neat Resin Samples At 700F And 4 Atmospheres In 100cc/MIn Dry Air

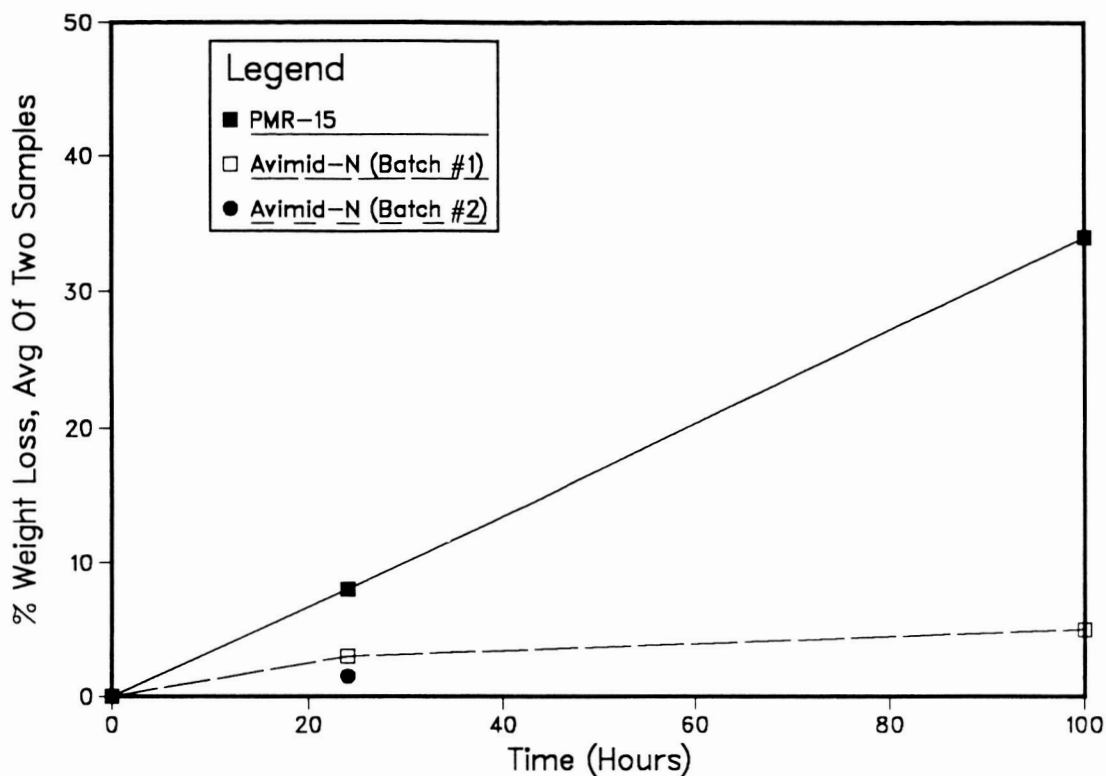


Figure 4. Comparison of Thermal-oxidative Stability PMR-15 and Avimid N



Figure 5. Avimid N Ring Subelement

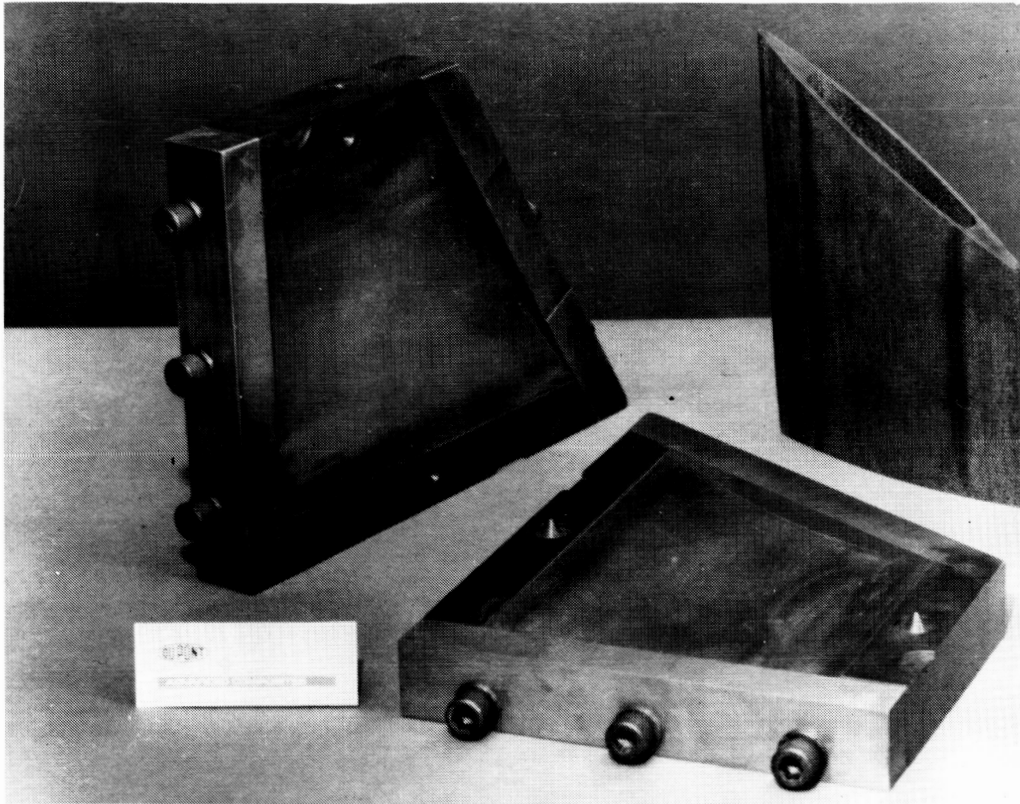


Figure 6. Strut Subelement Tool and Cured Strut

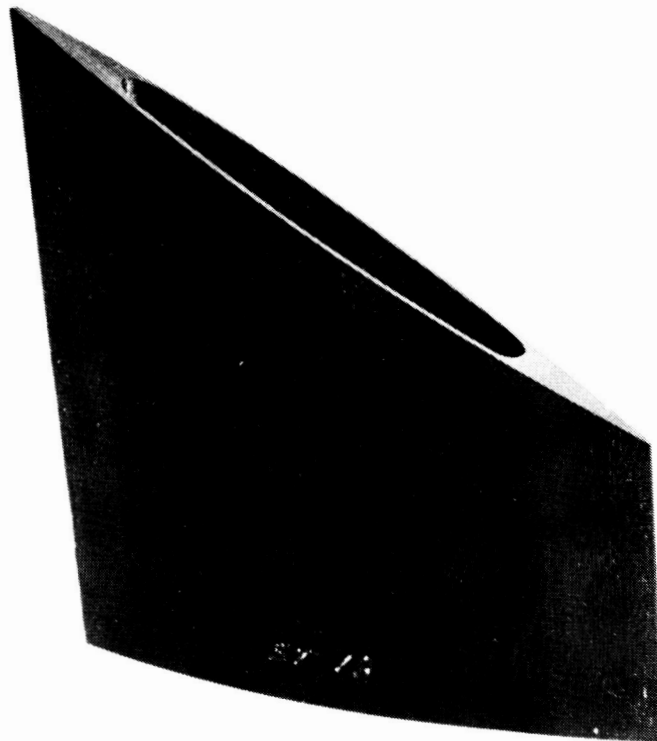


Figure 7. Strut with Good Consolidation

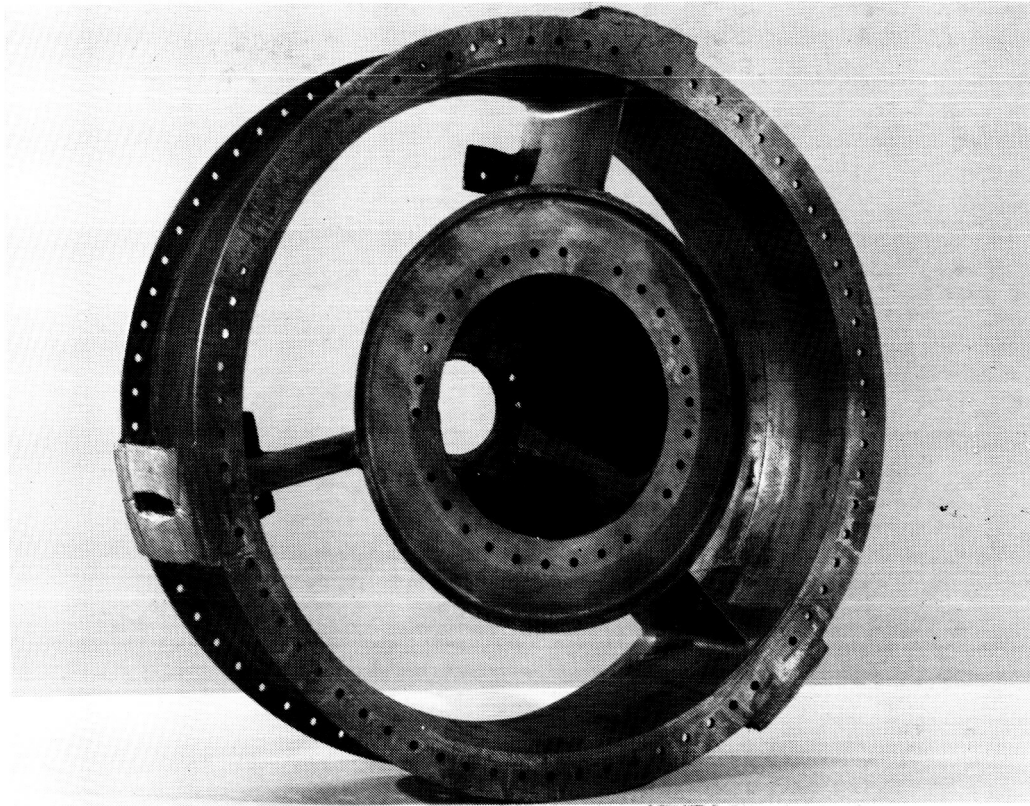


Figure 8. Front Side of Composite Half-Scale Case Subelement

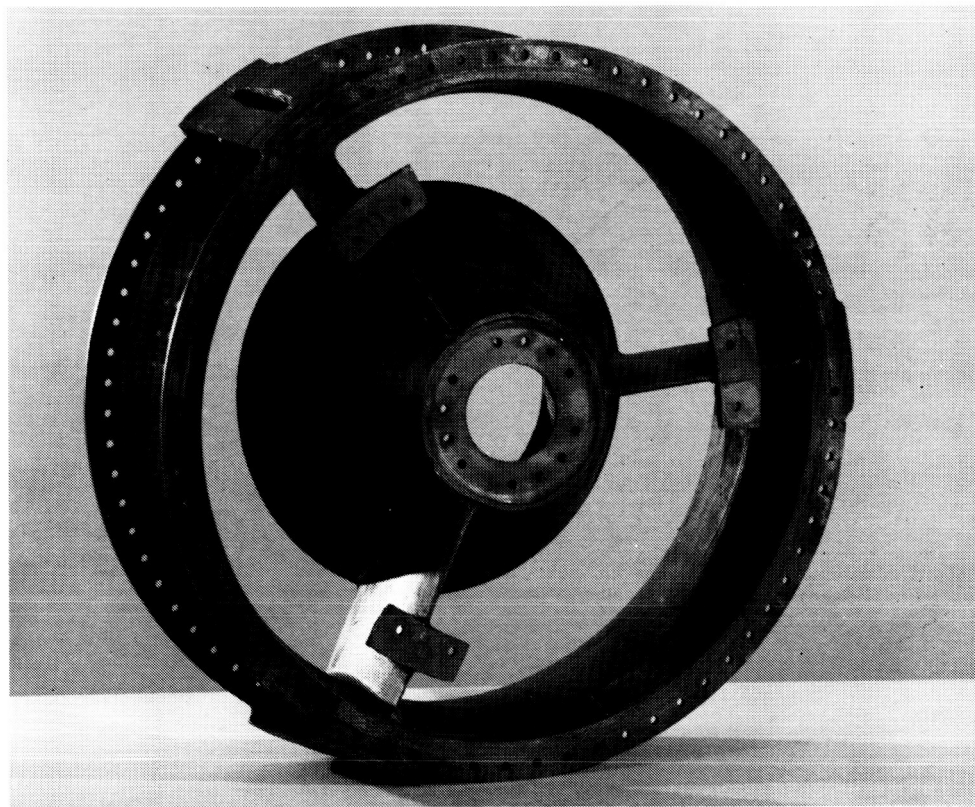


Figure 9. Aft Side of Composite Half-Scale Case Subelement



HALF SCALE METAL  
FLOW PATH TOOL

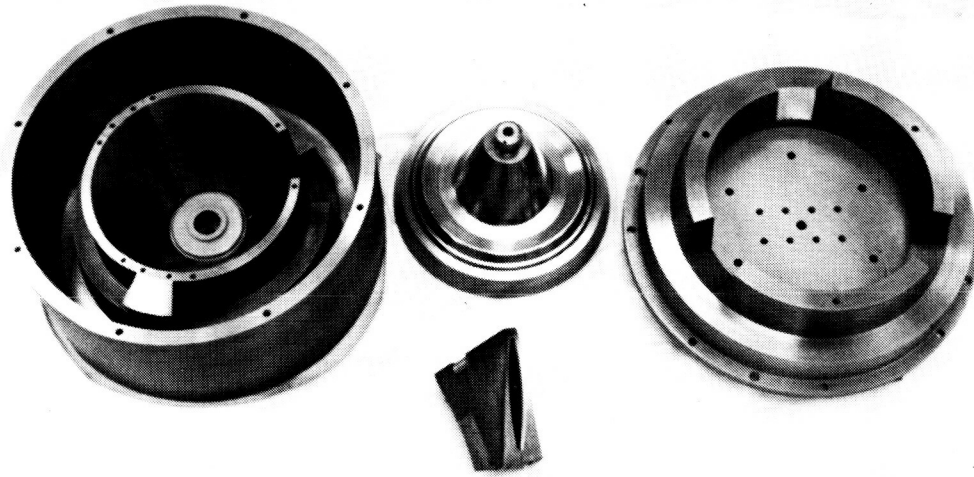


Figure 10. Mold for Half-Scale Case Subelement

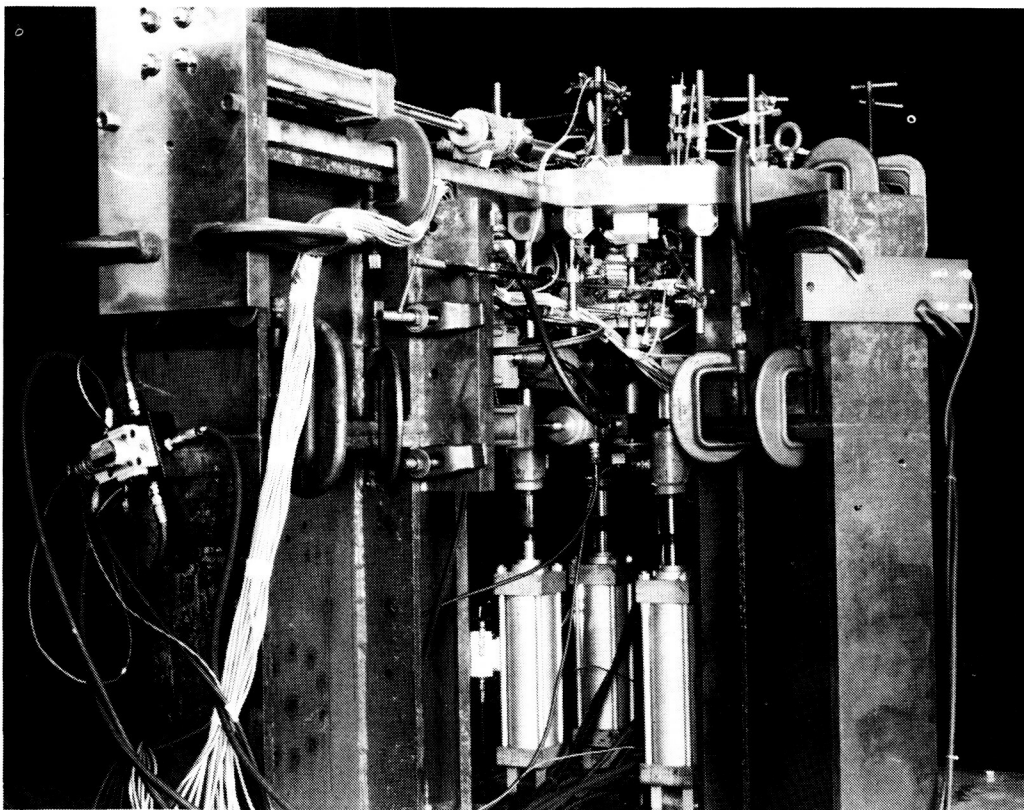


Figure 11. Test Rig for Structural Testing of Composite Subelements

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RESIN TRANSFER MOLDING OF TEXTILE PREFORMS FOR  
AIRCRAFT STRUCTURAL APPLICATIONS

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## INTRODUCTION

The NASA Langley Research Center is conducting and supporting research to develop cost-effective fabrication methods that are applicable to primary composite aircraft structures. One of the most promising fabrication methods that has evolved is resin transfer molding (RTM) of dry textile material forms. RTM has been used for many years for secondary structures, but has received increased emphasis because it is an excellent method for applying resin to damage-tolerant textile preforms at low cost. Textile preforms based on processes such as weaving, braiding, knitting, stitching, and combinations of these have been shown to offer significant improvements in damage tolerance compared to laminated tape composites. The use of low-cost resins combined with textile preforms could provide a major breakthrough in achieving cost-effective composite aircraft structures. RTM uses resin in its lowest cost form, and storage and spoilage costs are minimal. Near net shape textile preforms are expected to be cost-effective because automated machines can be used to produce the preforms, post-cure operations such as machining and fastening are minimized, and material scrap rate may be reduced in comparison with traditional prepreg molding.

Successful RTM is dependent upon many factors, including tooling approaches, resin characteristics, and textile preform architecture. Location of resin injection ports in the mold, resin viscosity variation with time and temperature, and compaction and permeability characteristics of preforms are all important factors that must be understood. Many of the RTM processes used in the past were developed using trial and error methods. Low fiber volume fraction (50 percent or less) secondary structures using fiberglass mats are readily molded by RTM. However, aircraft-quality primary structures have stringent structural requirements and processing conditions, necessitating a science-based approach to RTM process development. The purpose of this paper is to discuss experimental and analytical techniques that are under development at NASA Langley to aid the engineer in developing RTM processes for airframe structural elements. Included are experimental techniques to characterize preform and resin behavior and analytical methods that have been developed to predict resin flow and cure kinetics.

The NASA Langley RTM development team includes in-house staff devoted to process development, element fabrication, and composite material characterization; Virginia Polytechnic Institute and State University performing preform characterization and analytical modeling; the College of William and Mary developing resin characterization data and sensor systems; and industrial contractors (Douglas, Boeing, Lockheed, and Grumman) involved in process development, tooling studies, and subcomponent fabrication.

## RESIN TRANSFER MOLDING

The term RTM is often used rather loosely to include several processes such as vacuum/pressure infiltration, resin film infusion, and liquid pressure injection. In all cases, liquid resin infiltrates a dry preform, air is evacuated, and heat is applied to cure the composite. Several different types of tooling concepts have been developed to accommodate RTM variations. The preform may be placed in a matched cavity mold or on a single-sided mold and covered with a flexible membrane. Based on the characteristics of the resin (liquid or solid at room temperature, short or long gel time), infiltration may occur isothermally or during heat-up. Pressure used to compact the preform may also drive the flow of resin, or the compaction and injection pressures may be controlled separately.

Figure 1 illustrates the process (vacuum/pressure infiltration) used at NASA Langley to fabricate panels from preforms having through-the-thickness reinforcement using hot melt epoxy

resins. Flow occurs through the thickness of the preform as the resin melts and pressure is applied in a heated press. A tight fit is required between the preform and mold cavity to avoid a resin leak path around the preform. The pressure performs two functions: debulking the preform, and forcing the resin into the preform.

Figure 2 shows a typical matched cavity process employing pressure injection of resin. Flow occurs in the plane of the preform. Resin is contained in an external tank, which may be heated to lower the viscosity. Compaction pressure is used to close the mold and debulk the preform and is independent of resin injection pressure.

Some versions of RTM employ a combination of features of the two processes shown, leading to many variations of mold construction and process parameters. Analytical methods are needed to aid the mold designer and process engineer to minimize the waste of time and material. Successful modeling requires data on preform and resin processing properties; however, most of these data are not supplied by the preform or resin manufacturers. NASA Langley has sponsored the development of characterization methods for constituent materials and analytical models describing the RTM process. Recent results of this research are discussed herein.

### Preform Behavior

Textile preforms are deformable and porous in their dry state. The amount of rigidity and permeability is related to the fiber architecture. Several preform architectures are illustrated in figure 3. Another factor affecting preform rigidity and permeability is the size and type of the constituent fibers. High modulus, brittle fibers impose limits on the amount of crimp and tightness of the preform as it is produced. When compressed in the thickness direction, preforms exhibit nonlinear behavior as shown in figure 4. The reasons for the nonlinearity are that fibers deform by bending, behaving as constrained beams, by buckling, behaving as constrained columns, and also slide past each other with variable friction effects.

Compaction characteristics of preforms are quantified by mounting a sample between rigid plates, applying a compaction load and measuring the resulting thickness as shown in figure 4. Data are plotted in terms of fiber volume fraction, which is directly related to thickness, against pressure in figure 5. As indicated in the figure, none of the preforms meets the typical goal of 60 percent fiber volume in the uncompacted condition. Stitching helps to compact the Hexcel<sup>1</sup> preform, but it still falls short of the goal. In all cases shown, pressure in excess of vacuum (14.7 psi) must be applied to achieve the desired compaction.

Permeability is also a nonlinear function of preform compaction pressure. Pores exist between fiber bundles (tows) and at the tow intersections in the preform. If a preform is visualized as a collection of planar pores, the pressure drop at a fixed volumetric flow rate is proportional to the third power of pore height (ref. 1), which is proportional to fiber volume fraction. In addition, pores may change shape or close off entirely as compaction pressure and fiber volume fraction increase. Pores formed by aligned tows form a fairly smooth flow path, whereas pores formed by crossed tows create a much more tortuous path for resin flow as shown in figure 6.

<sup>1</sup>The use of trademarks of names of manufacturers in this paper does not constitute an official endorsement, either expressed or implied, of such products or manufacturers by the National Aeronautics and Space Administration.

Permeability is measured in a specially designed fixture as shown in figure 7. The sample is placed in a sealed chamber of variable height. The chamber is configured to allow flow in only one of the three primary directions at a time. Fluid of a known viscosity is pumped through the preform, and the resulting pressure drop is recorded. Permeability is calculated by multiplying flow rate, viscosity, and flow length and dividing by the product of flow pressure and area. This test is repeated at several preform thicknesses. Results are shown in figure 8 for a stitched and unstitched quasi-isotropic uniweave preform. Two effects are significant: permeability is much higher in-plane than through-thickness, and the presence of stitches affects flow differently in the two directions. The stitches create channels through the thickness of the preform which increase permeability in that direction. However, they also form a partial barrier to flow in the plane of the preform, thereby reducing in-plane permeability.

Permeability information is useful in process development at a number of levels. Molds are sometimes designed so that infiltration occurs below the target fiber volume fraction of 60 percent so that permeability is high; then full compaction is applied mechanically prior to gelation to squeeze out excess resin. However, this procedure is difficult for complex shapes. Another technique is to take advantage of thermal expansion differences between mold components to achieve the same effect. For example, silicone blocks placed inside an aluminum mold will expand and compact the preform as temperature increases. Injection occurs at moderate temperature and low fiber volume fraction. The final fiber volume fraction is attained at the higher cure temperature.

Mold design criteria can be partly established by consideration of permeability variation over a range of fiber volume fractions. Mold cavity machining tolerances and rigidity determine the uniformity of permeability of the preform. Figure 9 shows the permeability variation of 16 plies of 8 harness satin, IM7 graphite fabric. The nominal cavity thickness required for a 60 percent fiber volume is 0.264 inches. If this dimension is allowed to vary by  $\pm 0.005$  inches, the fiber volume ranges from 59 to 61 percent. If the dimension varies by  $\pm 0.020$  inches, the fiber volume ranges from 55 to 65 percent. The corresponding permeabilities are affected to a far greater degree than the fiber volume fractions. The nominal permeability, in units of  $10^{-10} \text{ in}^2$ , is 6.0. Permeability variation with the small tolerance ranges from 5.0 to 7.4. However, the variation with the large tolerance is 2.0 to 15.6, a range of more than twice the nominal value, which will most likely cause problems during resin infiltration.

### Resin Characterization

Resins are primarily characterized by viscosity measurements under both isothermal and increasing temperature conditions. Since viscosity behavior is affected by the existing degree of resin cure, this parameter must also be quantified. The degree of cure  $\alpha$  of a resin varies with time and temperature and is measured by differential scanning calorimetry (DSC), which records heat of reaction during time. This test is run under both isothermal and rising temperature conditions.

Resin viscosity can range from 10 cp to  $10^{15}$  cp before molecular forces restrict fluidity and elastic behavior sets in (ref. 2). Mold design features and process parameters must be selected to account for this behavior so that a fully impregnated void-free structure is produced. Examples of viscosity behavior are shown in figure 10, which shows viscosity as a function of time for two epoxies, each at two selected temperatures. An important point is that these data are for freshly formulated resins with no impurities; viscosity may be drastically affected by improper storage or formulation, and its dependence on time at temperature can vary substantially.

For RTM, it is desirable to tailor a mold and process window for minimum viscosity for the longest time. A relatively viscous resin such as 3501-6 (see figure 10) is recommended for a through-thickness RTM process only. A resin with low viscosity and short gel time such as

E905L can be used for in-plane injection, but would require multiple inlet ports to rapidly fill large molds. The E905L has acceptable pot life as long as the temperature is carefully controlled. Other resins that have been evaluated by NASA Langley are listed in figure 11. Difficulties arise when a process is optimized for a specific resin, but a decision is made to alter the resin formulation or switch to an entirely new resin. Such changes may not require mold redesign in the case of a prepreg material, but may require an existing RTM mold to be discarded in favor of a new design. Adjusting the process parameters may, however, in some cases allow continued use of the existing mold. These examples highlight the need for a predictive model of resin behavior if high-quality, defect-free composites are to be fabricated.

### Tackifiers

In some cases, textile processes yield a nearly net-shaped preform, such as braiding directly over a mandrel which is part of a RTM mold. However, in most cases, the textile product is supplied on a roll. The composite fabricator unrolls the material and cuts patterns that are assembled into the final three-dimensional preform. For complex shapes, the assembly process must be aided either mechanically, such as by stitching, or by bonding with tackifying agents as described below. Figure 12 illustrates a preform for a sine spar. Several plies of a multiaxial warp knit fabric were assembled using a tackifier and a shaping mold.

Another reason for using these tackifier preforming aids is to debulk the preform. As shown previously, even high density stitching does not fully debulk preforms. Tackifiers applied only to surfaces of plies do not fully debulk the entire thickness, as shown schematically in figure 13. The only ways to fully debulk a preform are by distributing a tackifier through the entire thickness (as by using powder-coated tows for the weaving process), by maintaining in-plane tension (as when braiding over a mandrel), or upon closing the mold. Considerable effort sometimes is needed to devise methods of closing molds without pinching or wrinkling preforms. The proper use of a tackifier would simplify the mold design.

Several types of tackifiers are listed in figure 13. They are available in a wide range of application temperatures and physical forms, from room temperature sprays to higher temperature scrims. Some are thermoplastics that remain as a discrete phase in the composite after cure, while others are formulated to dissolve in the matrix resin during infiltration. In any case, two criteria must be met: they must have minimal impact on resin infiltration, and they must have minimal impact on mechanical properties.

The effects of one type of tackifier (a polyamide scrim interleaved between each ply) on processing and mechanical properties of an eight harness satin fabric are shown in figure 14. Both the compaction pressure and the permeability at 60 percent fiber volume are very similar both with and without the tackifier. There is also little difference in room temperature static compression strength.

## SCIENCE-BASED RTM PROCESSING

### RTM Process Modeling

The approach in developing a process model for RTM has been to derive expressions for one-dimensional flow including all relevant physical effects, verify the model experimentally, then expand to complex geometries.

The flow of fluid through porous media obeys Darcy's law:

$$Q = \left( \frac{K}{\mu} \right) \frac{\Delta P}{X} A \quad (1)$$

where  $Q$  is the volumetric flow rate,  $K$  is the preform permeability,  $\mu$  is the resin viscosity,  $\Delta P$  is the pressure difference between the resin source and the flow front over a distance  $X$ , and  $A$  is the area normal to the direction of flow. This is a simple equation, but complexity arises because  $K$  varies with fiber volume fraction  $V_F$  and fiber orientation;  $\mu$  varies with temperature  $T$ , time  $t$  and degree of cure  $\alpha$ ; and complex geometries include the  $Y$  and  $Z$  dimensions in which  $A$  may also vary.

For a simple isothermal one-dimensional case with a nonreacting resin and homogeneous incompressible preform, infiltration pressure and mold filling time can be calculated by hand using equation (1). However, the vacuum/pressure through-thickness process involves the simultaneous interaction of all of the parameters in both time and space. A computer is required to expedite the solution.

A computer model of the RTM process has been under development at Virginia Polytechnic Institute and State University. Basic features of this model are shown schematically in figure 15. The user specifies the process conditions and mold configuration. The main program calls up subroutines that contain data for the specific preform and resin. The program outputs values of temperature, viscosity, position of flow front, degree of cure, fiber volume fraction, and thickness.

Details concerning the calculations performed in the main program are described in reference 3. Compaction behavior is modeled with a logarithmic expansion which requires four constants to be determined experimentally. Permeability is related to fiber volume fraction using the Kozeny-Carman or Gebart equations, each of which requires an experimentally-determined constant. Resin viscosity is modeled using either an exponential or a William-Landel-Ferry expression, which relate viscosity to temperature and degree of cure. Material properties for these expressions are determined from viscosity and DSC tests. The degree of cure of a given resin is related to heat of reaction and can be modeled with exponential equations relating time and temperature history. The form of these equations can be very different for various resins, and considerable judgement and analysis of DSC information are required for accurate modeling. Once the forms of the degree of cure equations are selected, several material constants must be determined from DSC data.

Transient conductive heat flow is a major factor in the through-the-thickness process with hot melt resins. The heat flow model includes the change in thermal conductivity as resin infiltrates the preform and accounts for heat generated by the resin cure reaction. Convective heat transfer between the preform and slowly moving resin is neglected. Material thermal properties are taken from published sources. Heat generation from the resin reaction is determined from the degree of cure calculation.

The flow front position is calculated using a finite element procedure in discrete time increments. Darcy's Law is applied to each element between the resin source and the flow front, using the instantaneous values of permeability and viscosity for each element.

Computing requirements are related to the dimensions of the problem as indicated in figure 15. The one-dimensional model will run on a personal computer in 1/2 hour or less. However, the two- and three-dimensional models require larger computers and more time. The added complexity of higher order models arises because of the orthotropic permeability of preforms and the need to calculate flow front position in multiple directions.

## RTM Process Monitoring and Control

The College of William and Mary has been developing a sensing system for the cure state of resins based on dielectric measurements. This system, termed Frequency-Dependent Electromagnetic Sensing (FDEMS), figure 16, utilizes a thin (0.003 inch) sensor consisting of interdigitated electrodes mounted on a substrate. The sensor is excited by an alternating electric current at various frequencies, which in turn causes molecular vibration in the resin. The molecular action changes as the reaction proceeds, which alters the electrical conductivity and capacitance of the resin. FDEMS correlates electrical conductivity and capacitance to resin viscosity and degree of cure. This system can be applied to any of the RTM process variations to detect the presence or absence of resin, local viscosity, and degree of cure.

Another use of FDEMS is to enable process control based on sensing the direct process variables (viscosity, degree of cure) in real time, as opposed to the traditional method of controlling indirect variables (mold temperature, time) per a predefined cycle. This procedure is illustrated in figure 17. Panels have been made with the through-the-thickness infusion process under direct control of temperature by the FDEMS system. The system was programmed to maintain a preheat temperature until both bottom and top sensors indicated wet-out, then to proceed with the cure cycle.

The use of the FDEMS system in a control mode is being expanded to include larger parts with more complex flow paths, emerging resins, and control of pressure cycles.

## Model Verification and Utilization

The one-dimensional model has been verified using a through-the-thickness infusion process with three different preforms, 3501-6 resin, and several cure cycles (ref. 3). The mold was instrumented with thermocouples for determining temperature distribution, a dial gage to measure platen deflection for correlation with flow front position, and the FDEMS system to monitor viscosity and degree of cure, and to verify complete infiltration. Results for one case are shown in figure 18. Agreement is very good for temperature and flow front calculations; predictions of viscosity and degree of cure, although showing some disagreement with experimental observations, are still reasonable.

The model was used to derive alternate cure cycles for panels of 0.25 inch nominal thickness as shown in figure 19. The resin manufacturer recommended a moderate temperature hold during infiltration, cycle A. The model was run with the manufacturer's and two other heating cycles: ramping to the cure temperature, cycle B, and preheating the platens, cycle C. Predicted infiltration times agree well with experimentally determined values, with cycle C reducing infiltration time by 50 percent.

A further example of the utility of the model is in deciding whether to use or discard an aged sample of resin. The degree of cure advancement during storage,  $\alpha_0$ , is determined from a DSC test. For example, hypothetical values of 0.02 and 0.30 were entered in the model, which was run with one of the above cure cycles. The case with aged resin predicts infiltration into only about 1/3 of the full thickness before gelation occurs, whereas full infiltration was predicted for fresh resin. The reason for the difference can be explained by plotting the reciprocal of viscosity against time for the resin flow front, as in figure 20. The areas under the curves are related to the process window. For fresh resin, the area is .00900 min/cp, whereas for the aged resin, the area is only



.00045 min/cp. Further model runs with aged resin and fewer plies indicate more complete infiltration, with saturation occurring at 0.063 inch thickness. If this is not thick enough for the user, the resin should be discarded.

## SUMMARY

Resin transfer molding (RTM) is a promising method for cost-effective fabrication of composite structures having a wide range of preform architectures and resin processing requirements. The large range of variations of the process and of material behavior requires that a science-based understanding be applied to the design of molds and the development of cure cycles.

Characterization tests on textile preforms have shown that compaction pressure in excess of ambient vacuum is required to achieve a 60 percent fiber volume. Permeability of preforms can differ by at least two orders of magnitude between the in-plane and through-thickness directions. The effects of secondary materials such as stitches and tackifiers on compaction and permeability should also be quantified.

Resin flow and cure kinetics have been successfully modeled mathematically. Flow simulations can be performed on a computer in order to save labor and materials during process development. Frequency-dependent dielectric sensors have been shown to be beneficial in both monitoring and controlling the RTM process.

Research supported by NASA Langley in developing the required process models, databases, and sensing systems is beginning to yield solutions to practical problems. The work is currently being expanded to encompass a wider range of geometries and materials.

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2. Van Vlack, L. H.: Materials Science for Engineers, Addison-Wesley Publishing Co., Inc., 1970, p. 232.
3. Weideman, M. H.: An Infiltration/Cure Model for Manufacture of Fabric Composites by the Resin Infusion Process, Masters Thesis, Virginia Polytechnic Institute and State University, Blacksburg, Virginia, 1991.

- 1-Dimensional flow with hot melt resins
- Compaction, infiltration and heat transfer are coupled

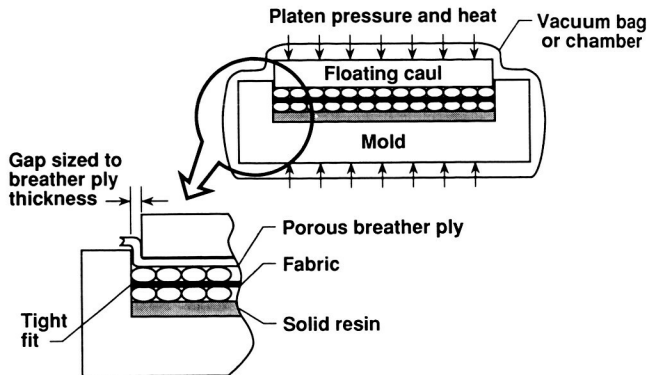


Figure 1. Through-Thickness Vacuum/Pressure Process

- 2-Dimensional flow
- Compaction and infiltration are decoupled
- Resin and mold are normally preheated

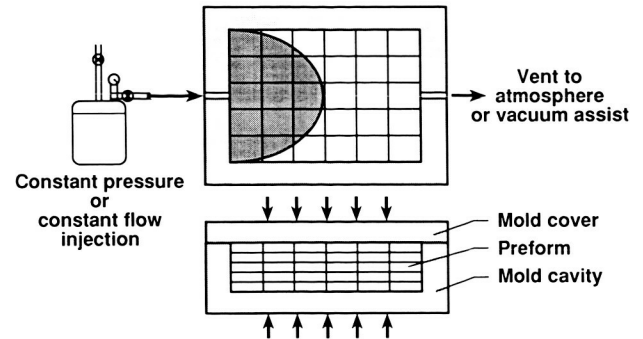


Figure 2. Pressure Injection

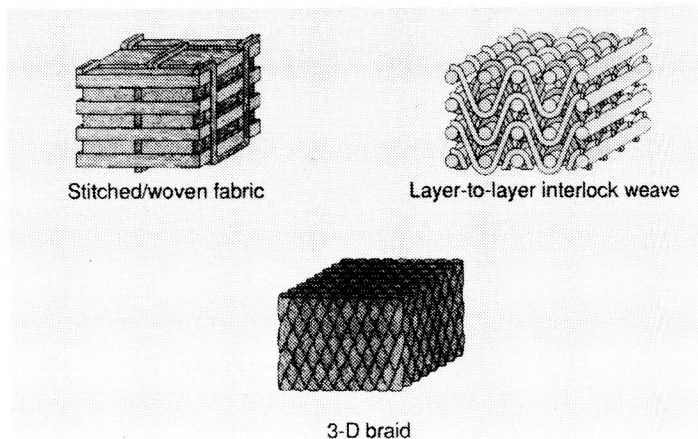


Figure 3. Through the Thickness Reinforced Textile Forms

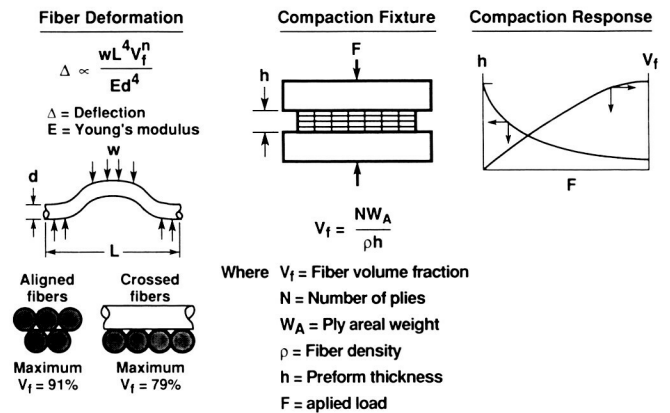


Figure 4. Preform Compaction Behavior

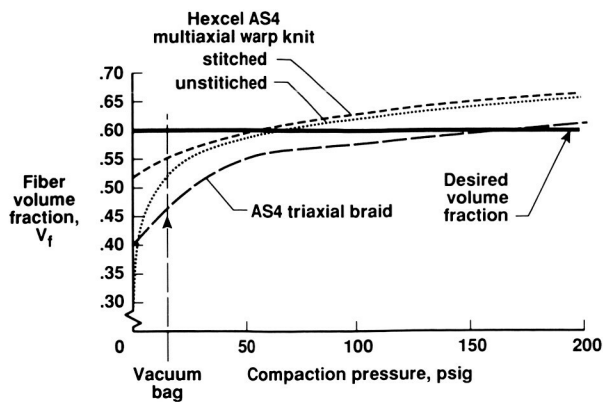


Figure 5. Preform Compaction Data

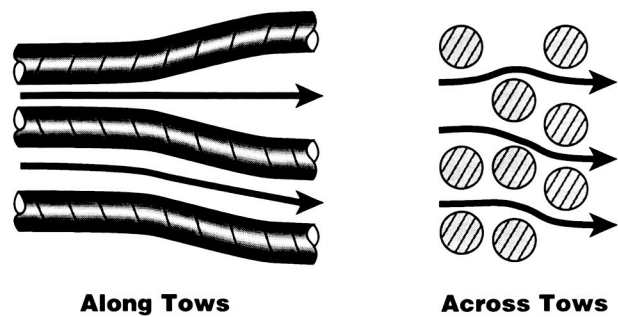


Figure 6. Flow Through Pores in Fibrous Preforms

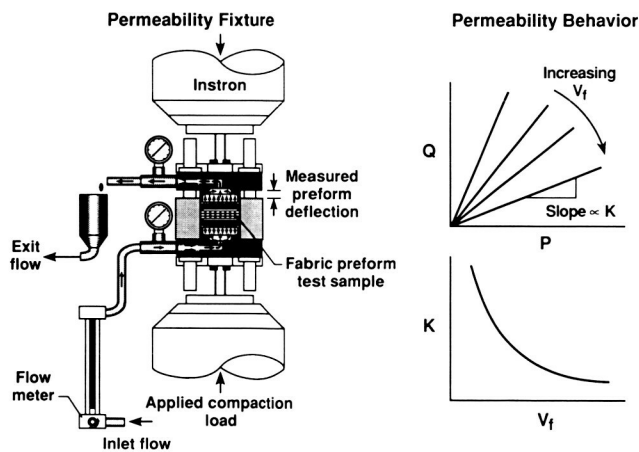


Figure 7. Preform Permeability Measurement

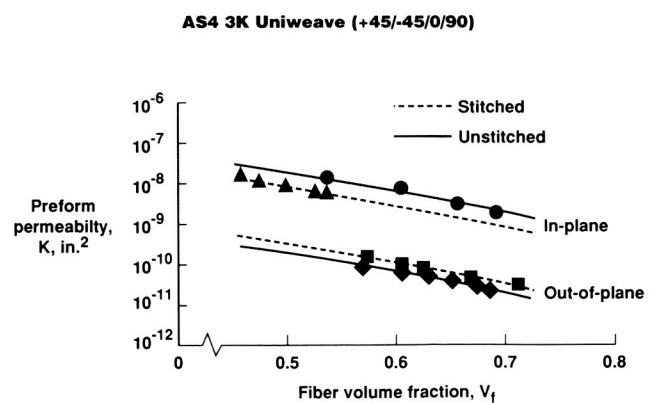


Figure 8. Preform Permeability Data

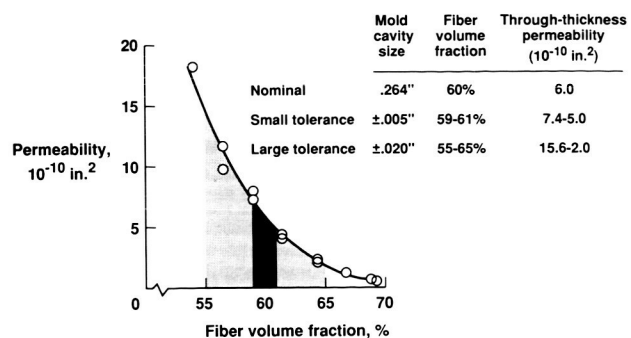


Figure 9. Permeability Variation of Satin Fabric

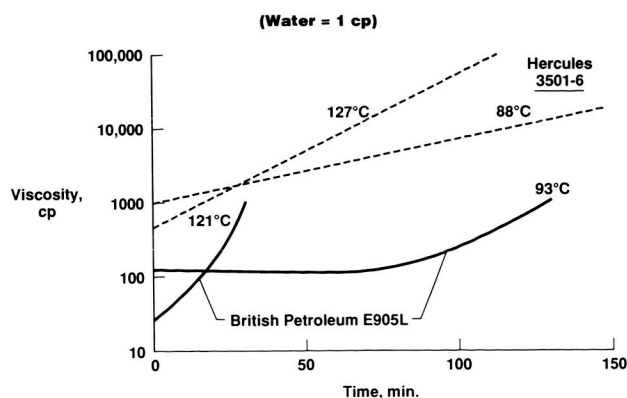


Figure 10 Effect of Time and Temperature on Viscosity of Epoxy Resins

|          |                |          |
|----------|----------------|----------|
| Hercules | 3501-6         | Hot melt |
| Dow      | CET 2          | Hot melt |
| 3M       | PR500          | Paste    |
| Dow      | Tactix 138/H41 | Liquid   |
| BP       | E905L          | Liquid   |
| Shell    | 1895/W         | Liquid   |
| Shell    | 862/763        | Liquid   |

Figure 11. Resins Evaluated

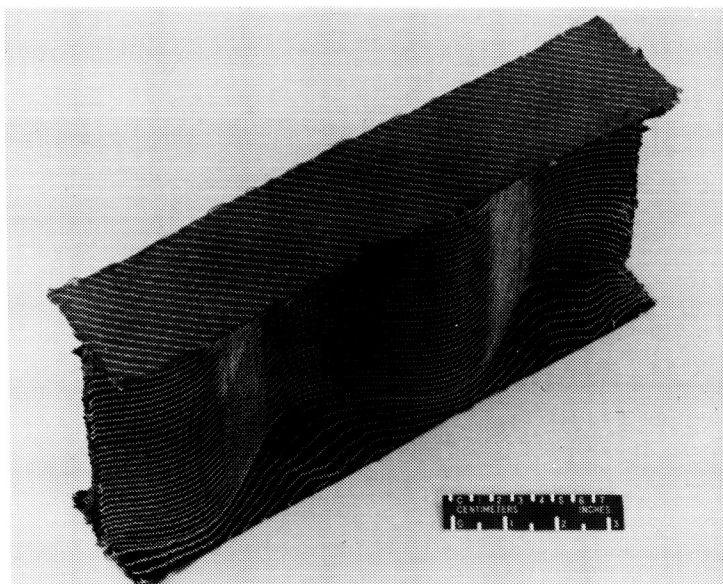


Figure 12. Preform for Sine Wave Spar

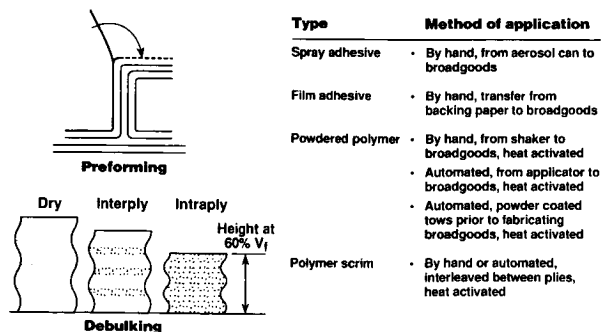


Figure 13. Tackifier Materials

(16 plies, 8 harness satin, 0/90, IM7  
15 plies polyamide scrim, interleaved)

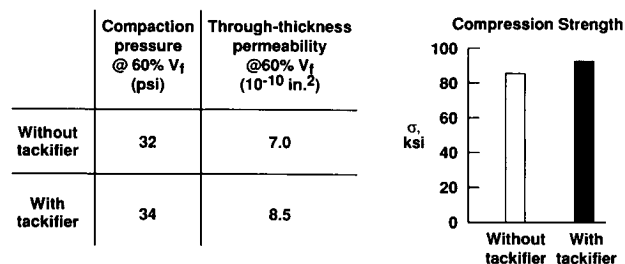


Figure 14. Effect of Tackifier on Processing and Properties

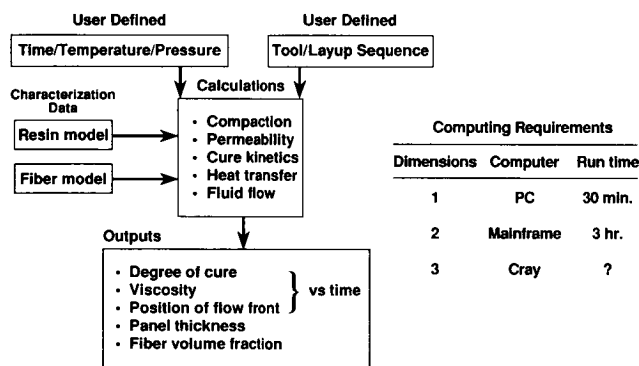


Figure 15. Computer Simulation of RTM Processes

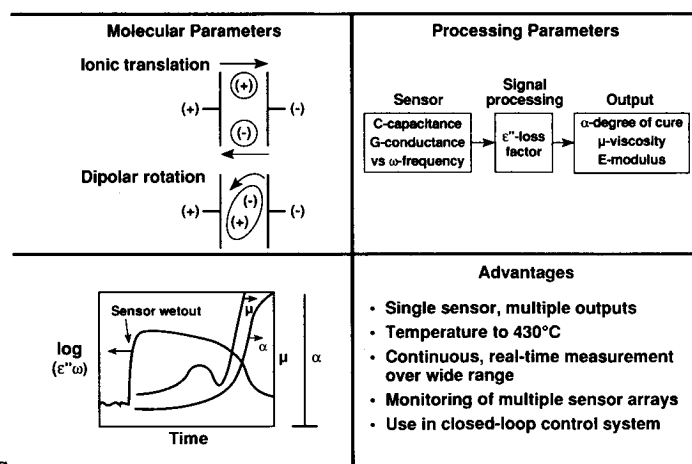


Figure 16. Frequency-Dependent Electromagnetic Sensors (FDEMS)

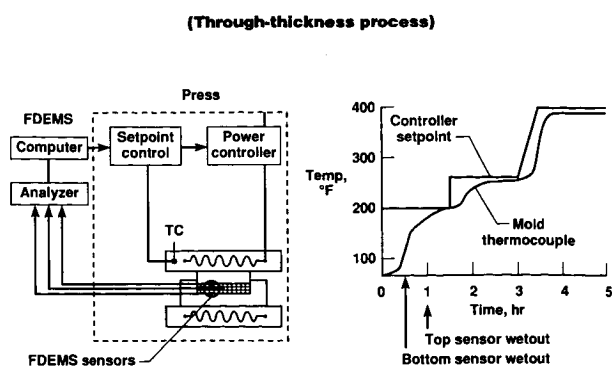


Figure 17. Prototype FDEMS Expert Cure System

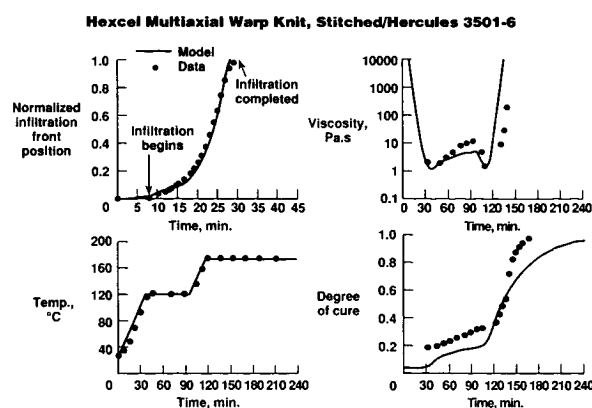


Figure 18. 1-D Model Verification

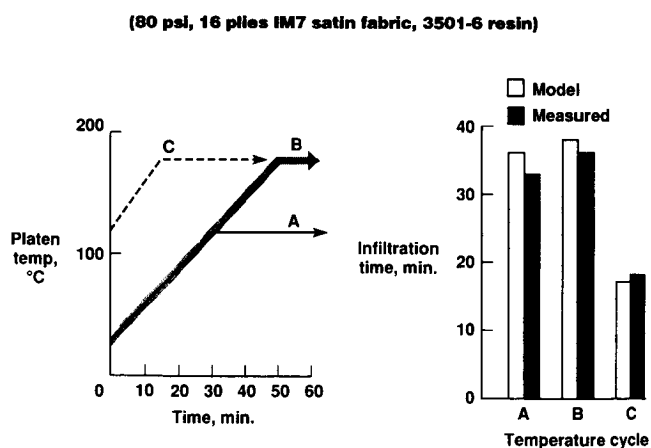


Figure 19. Utility of 1-D Model to Alter Infiltration Time

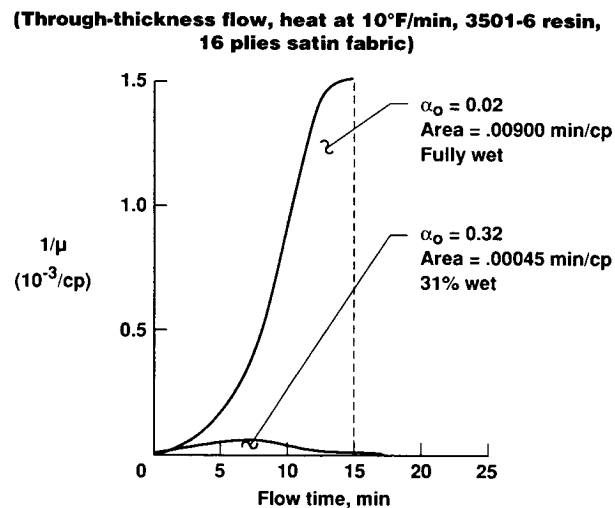


Figure 20. Use of RTM Flow Model to Define Process Window

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**SESSION VIII**  
**DESIGN APPLICATIONS (B)**



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## ANALYSIS OF AIRCRAFT ENGINE BLADE SUBJECT TO ICE IMPACT

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## ABSTRACT

The ice impact problem on the engine blade made of layered composite is simulated. The ice piece is modeled as an equivalent spherical object and has the velocity opposite to that of the aircraft with direction parallel to the engine axis. Near the impact region and along the leading edge, the blade is assumed to be fully stressed and undergoes large deflection. A specified portion of the blade around the impact region is modeled. The effect of ice size and velocity on the average leading edge strain are investigated for a modified SR-2 model unswept composite propfan blade. Parametric studies are performed to study the response due to ice impact at various locations along the span. Also, the effects of engine speed on the strain and impact displacements are discussed. It is found that for a given engine speed, a critical ice speed exists that corresponds to the maximum strain and this critical speed increases with increase in the engine speed.

## INTRODUCTION

At high altitudes, when aircraft flies through clouds of super-cooled water droplets, ice formation occurs on forward facing structural components. One such component is the engine inlet. With time, the ice accretes on the inlet and eventually sheds due to structural vibrations. A schematic of this phenomenon is shown in Figure 1. As a result, blocks of ice travelling at the speed of aircraft impacts the engine blades rotating at high RPM. This process may cause severe damage to the blade and subsequently to the engine. In order for the blade to sustain the ice impact, it is necessary to properly account for these constraints during the design.

Fibrous composites are ideal for structural applications such as high performance aircraft engine blades where high strength-to-weight and stiffness-to-weight ratios are required. These factors along with the flexibility to select the composite layup and to favorably orient fiber directions help limit the impact damage and stresses arising from large rotational speeds.

The objective of this paper is to simulate ice impact on an engine blade. SR-2 unswept composite propfan blade is considered for local damage analysis. The impact analysis is carried out by modifying the foreign object damage option provided in Reference [1]. It is assumed that the damage is severe in the local region and hence only a specified portion of the blade around the impact region is modeled. Furthermore, large scale deflections accompany the impact event in the impact zone [2]. In order to simulate this behavior near the impact region and along the leading edge, the blade is assumed to be fully stressed and undergoes large deflection. This is accomplished in the code by modifying the membrane stiffness in the radial direction to reflect a fully yielded condition. A detailed justification for these assumptions is also given in Reference [2]. Parametric studies are performed to study the response of the blade due to ice impact at various locations along the span. The effect of ice size and velocity on the average leading edge strain and maximum impact displacements are investigated.

## IMPACT ANALYSIS

### Geometry of Ice Impact

The geometry of ice impact on the leading edge of the blade is shown in Figures 2(a,b). Impact velocity direction relative to the blade is a function of aircraft speed and the rotational speed of the blade (Figure 2a). The resulting impact force not only depends on the magnitude and direction of relative velocity but also on the mass of ice piece. Depending on the spacing (i.e., number) of blades and relative velocity as shown in Figure 2b, only a portion of the ice piece hits the blade. The ice piece that is approaching the blade under consideration is sheared off by the adjacent blade and only a part of it impacts the leading edge. Effectively, the size of the ice piece that finally impacts the blade depends on the blade spacing, blade speed and aircraft speed.

### Modeling of the Blade

The damage due to impact is considered to be highly localized and hence only a portion of the blade around the impact region (i. e., a specified portion along the span and half of the blade along the chord as shown in Figure 3) is modeled. In the code, the blade geometry is input in the form of finite element grid and nodal thicknesses. The spanwise impact region is specified with two parameters; namely, lower and upper bounds of radial fractions,  $a$  and  $b$ . However, the modeled impact span is approximated to be the region between the two finite element radial stations that are defined in the blade geometry and are nearest to  $a$  and  $b$ . Along the chord, only half of the blade is considered for impact analysis. The finite element used is similar to the NASTRAN (TRIA3) three node triangular plate element [3,4]. A total of 35 nodes and 48 elements are used (Figure 3). The impact is considered at the midpoint of the local patch (finite element node 16) along the leading edge. All the edges except the leading edge are considered fixed as the impact response beyond these boundaries is negligible.

Large scale deflections usually accompany the impact event. This phenomenon is simulated in the elements that are close to the impact node by modifying the membrane stiffness in the radial direction (to reflect fully yielded condition) and zeroing-out the spanwise bending stiffness. These special elements are shown as shaded in the Figure 3. If  $\sigma_{\text{yield}}$  is the effective yield stress of the element in the radial direction;  $w_1$ ,  $w_2$  and  $w_3$  are out of plane displacements at 3 nodes; the modified stiffness associated with these degrees of freedom is given by [2],

$$[K] = \frac{y_3}{2x_2} \sigma_{\text{yield}} t_e \begin{bmatrix} 1 & -1 & 0 \\ -1 & 1 & 0 \\ 0 & 0 & 0 \end{bmatrix} \quad (1)$$

where  $t_e$  is the element thickness and  $x_2$  and  $y_3$  are the element local coordinates (Figure 4). The terms of this stiffness matrix replace the spanwise bending stiffnesses of the linear elastic triangle. The stiffness in Equation (1) reflects the perfectly plastic condition with constant stress ( $\sigma_{\text{yield}}$ ) in the radial direction.

Figure 5 shows the material configuration of the composite blade. In each finite element, the skin (material 1) is always present. The remaining materials exist only if the blade thickness permits them. Materials  $(n-1)$ ,  $(n-2)$  ... are dropped from the layup of an element in that order. If the element thickness can accommodate all the materials then the core ( $n^{\text{th}}$  material with variable thickness) is used to fill the remaining excess thickness.

#### Modeling of Ice Impact Loading

The impact force on the blade arises from the momentum of the ice piece. The temporal and spatial distribution of this impact force is carried out with the following considerations [2]:

- Only the impact force component normal to the blade chord will cause the local deformation.
- Load duration,  $T$ , of impact is the time taken by the equivalent spherical ice piece (radius,  $r$  and relative velocity,  $V_{\text{rel}}$ ) to squash-up, i. e.,  $T = 2r/V_{\text{rel}}$ .
- Impact force is distributed using a parabolic distribution function over all finite element nodes within one diameter (equivalent) of the ice piece from the impact node (see Figures 6a and 6b).

$$f_i = \frac{f_{av} [1 - x_i^2/4r^2] [1 - y_i/2r]}{\sum_{i \in \Pi} [1 - x_i^2/4r^2] [1 - y_i/2r]} \quad (2)$$

where  $f_i$  is the nodal force,  $f_{av}$  is the component of the impact force normal to the blade chord, and  $x_i$  and  $y_i$  are the local coordinates with impact node as the origin (Figure 6a).

- d) The impact wave travels along the chord at the same velocity as that of ice impact. As a result,  $i^{\text{th}}$  nodal loading start time,  $T_i$ , is defined by the ratio of nodal chordwise distance,  $d$  from the leading edge and the ice relative velocity,  $V_{rel}$ , i.e.,  $T_i = d/V_{rel}$ , (see Figure 6a).
- e) Blade flexibility reduces the instantaneous ice impact loads due to reduction in the relative velocity. However, deflection of the blade results in ingestion of extra ice and therefore, there is an increase in the total momentum exchange. These factors are accounted and nodal forces are adjusted accordingly (see Reference [2] for details).

### Transient Response Analysis

Transient response of the local patch is obtained by the process of modal integration. The modal analysis is based on the principle that statically deformed structure can be described by a combination of that structure's mode shapes. Once the coefficient of each mode's participation is calculated, the impact event can be simulated by the integration of a series of linear steps through time. The deflected shape due to ice impact is basic; therefore, only the first five modes of the local patch are used to compute the response. Also, damping is assumed to be zero as very little structural damping occurs during the impact event.

Upon diagonalizing the governing equations of motion of undamped system [5], the uncoupled equations with each modal coordinate,  $\xi_i$ , satisfies the second order differential equation,

$$\ddot{\xi}_i + \omega_i^2 \xi_i = \frac{1}{M_i} P_i(t) \quad (3)$$

where  $\omega_i^2 = K_i/M_i$ ,  $M_i$  and  $K_i$  are the  $i^{\text{th}}$  diagonal elements of the generalized mass and stiffness matrices, and  $P_i(t)$  is the  $i^{\text{th}}$  component of the generalized force at time  $t$ .

The general solution of the above equation, expressed in terms of arbitrary initial conditions ( $\xi_{i,n}$  and  $\dot{\xi}_{i,n}$  at  $t = t_n$ ), and a convolution integral of the applied load, can be written as,

$$\xi_i(t) = F(t-t_n)\xi_{i,n} + G(t-t_n)\dot{\xi}_{i,n} + \frac{1}{M_i} \int_{t_n}^t G(t-\tau) P_i(\tau) d\tau \quad (4)$$

where  $F$  and  $G$ , respectively are the solutions with unit displacement and unit velocity initial conditions.

Assuming that load varies linearly between  $t_n$  and  $t_{n+1}$ , the solution at time  $t_{n+1}$  can be obtained in closed form as (see Reference [6]),

$$\xi_{i,n+1} = F(h)\xi_{i,n} + G(h)\dot{\xi}_{i,n} + A P_{i,n} + B P_{i,n+1} \quad (5a)$$

$$\dot{\xi}_{i,n+1} = F'(h)\xi_{i,n} + G'(h)\dot{\xi}_{i,n} + A' P_{i,n} + B' P_{i,n+1} \quad (5b)$$

where  $h = t_{n+1} - t_n$ ,  $F(h) = \cos \omega_i h$ ;  $G(h) = (1/\omega_i) \sin \omega_i h$ ;  $F'(h) = -\omega_i \sin \omega_i h$  and  $G'(h) = \cos \omega_i h$  for the undamped system [5].

For zero initial conditions ( $\xi_{i,0} = \dot{\xi}_{i,0} = 0$ ), the coefficients  $A$ ,  $A'$ ,  $B$  and  $B'$  are given by

$$A = \frac{1}{M_i \omega_i^2} \left( \frac{\sin \omega_i h}{\omega_i h} - \cos \omega_i h \right), B = \frac{1}{M_i \omega_i^2} \left( 1 - \frac{\sin \omega_i h}{\omega_i h} \right) \quad (6a)$$

$$A' = \frac{1}{M_i \omega_i} \left( \sin \omega_i h - \frac{1}{\omega_i h} + \frac{\cos \omega_i h}{\omega_i h} \right), B' = \frac{1}{M_i \omega_i} \left( \frac{1}{\omega_i h} - \frac{\cos \omega_i h}{\omega_i h} \right) \quad (6b)$$

The response at all time steps can be obtained by applying equations for  $\xi_{i,n+1}$  and  $\dot{\xi}_{i,n+1}$  recursively. At each time step, the modal coordinates,  $\xi_{i,n}$ , can be transformed to nodal displacements using modal matrix. From these displacements, large radial strains,  $\epsilon_x$ , of the leading edge elements are computed with the relation [2],

$$\epsilon_x = \frac{1}{2} \left[ \frac{w_2 - w_1}{x_2} \right]^2 \quad (7)$$

The entire calculation proceeds very quickly, due to the uncoupled nature of the modal responses. Among the strains at all the time steps, the maximum is picked as a representative quantity for the local impact damage.

## RESULTS AND DISCUSSION

The SR-2 model unswept propfan blade [7] is chosen to study the ice impact and the planform of the blade is shown in Figure 7. The setting angle (orientation of the blade chord with respect to the plane of rotation at 75% span) is  $57^\circ$  and the number of blades are 8. In the present study, the composite material layup: Titanium skin/Graphite-Epoxy ( $\pm 45^\circ$ )/Titanium core is used (Figure 8). For these materials, the properties are given in Table 1. The effective yield stress for element stiffness modification is assumed to be 206 ksi.

### Effect of Ice Size and Ice Velocity

The impact in the region 50-90% of the span is known to cause severe damage to the blade [2]. Hence, lower and upper limits of the modeling region are taken as 50% and 90% blade span for the impact analysis. For this modeling region, the impact radius is at 9.55" (73.5% span). These data are presented for three engine speeds, 3000, 5000 and 8000 RPM in the form of contour plots (Figures 9a-c). For all the engine speeds, the maximum strain corresponds to  $r = 0.8"$ . However, the corresponding critical ice speed varies depending on the engine RPM. As an example, at 3000, 5000 and 8000 RPM the peaks are respectively at 100, 130 and 190 knots. The associated peak values of the strains are 1.66%, 7.61% and 14.22%. From these plots, a safe range of ice impact parameter; namely, ice speed and ice size that can cause a specified local damage to the blade can be determined. Once the designer determines the tolerable strain along the leading edge of the blade, the combination of ice size and speed are defined by the region to the left of the level curve labelled with that strain.

For the maximum ice size, a typical variation of strain with ice speed is shown in Figure 10. The strain is zero at ice speeds of 0 and 211 knots, but it reaches a maximum value at 100 knots. This is due to the fact that only impact force normal to the chord ( $f \sin \theta$ ) gives rise to this strain. When ice speed = 0, the impact force ( $f$ ) is zero and when ice speed = 211 knots, the impact angle ( $\theta$ ) is zero. Initially the value of  $\theta$  is equal to the stagger angle ( $55^\circ$  at the impact radius) and gradually decreases to  $0^\circ$  as the ice speed increases (Figure 10).

Figure 11 shows the the effect of impact radius on the strain. The strain is nearly zero when the impact takes place below 53% span (approx. 7.5" radius) but steeply increases as the ice impacts towards the blade tip. This is expected as the blade is relatively thin closer to the tip. Moreover, the increase in impact radius results in increase in the blade velocity and consequently in impact force .

### Effect of Engine Speed

With engine RPM as the parameter (1000-10,000 RPM), the strain plot shown in Figure 10 is repeated in Figure 12. It can be seen that the variation of strain is smooth for engine speeds up to 4000 and fluctuates for the higher speeds. However, the average trend remains the same for all RPM and the peak strain is larger for higher engine speed. The probable cause of these local fluctuations is due to discrete nature of the nodal force distribution, i.e., the finite element nodes which are in the force distribution boundary are suddenly loaded when the impact wave reaches them (Figure 6a).

Relative magnitudes of typical impact (out-of-plane) displacement distribution for three engine speeds (3000, 5000 and 8000 RPM) is shown in Figure 13. The data corresponds to ice speed of 200 knots and ice radius of 0.8". The displacements along the leading edge remain the same and gradually decreases to zero toward the center of the blade. The maximum displacement for 5000 RPM is higher than that for 8000 RPM as the analysis ice speed (200) does not correspond to the critical one in both the cases.

## CONCLUSIONS

The ice impact on an aircraft engine blade made of layered composite is simulated. From the results presented in this paper, the following conclusions can be drawn:

1. The largest ice piece results in maximum average leading edge strain.
2. For a given engine speed, a critical ice speed exists that corresponds to the maximum strain. This critical speed increases with increase in the engine speed.
3. The leading edge strain increases steeply with increase in impact radius.
4. The strains generated from this analysis can be effectively used for structural tailoring of engine blades.

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3. MacNeal, R. H.: A Simple Quadrilateral Shell Element. Computers and Structures, vol. 8, Pergamon Press, Great Britain, 1978, pp. 175-183.
4. MSC/NASTRAN, Version 65C, User's Manual. The MacNeal-Schwendler Corporation, 1987.
5. Thomson, W. T: Theory of Vibration with Applications. Prentice Hall, Inc., Englewood Cliffs, New Jersey, 1981.
6. MacNeal, R. H.: The NASTRAN Theoretical Manual. NASA SP- 221 (01), 1972.
7. Reddy, T. S. R.; and Kaza, K. R. V.: Analysis of an Unswept Propfan Blade With a Semiempirical Dynamic Stall Model. NASA TM 4083, 1989.



Table 1: Properties of the Blade Constituent Materials

| Material Type  | $E_{11}$<br>(psi)    | $E_{22}$<br>(psi)    | $G_{12}$<br>(psi)   | $\nu_{12}$ | $\rho$<br>(lb.sec <sup>2</sup> /in <sup>4</sup> ) | ply thickness<br>(inches) |
|----------------|----------------------|----------------------|---------------------|------------|---------------------------------------------------|---------------------------|
| Titanium (Ti6) | 16.5x10 <sup>6</sup> | 16.5x10 <sup>6</sup> | 6.4x10 <sup>6</sup> | 0.30       | 0.00044                                           | ----                      |
| Graphite-Epoxy | 32.0x10 <sup>6</sup> | 1.0x10 <sup>6</sup>  | 0.7x10 <sup>6</sup> | 0.25       | 0.00015                                           | 0.005                     |

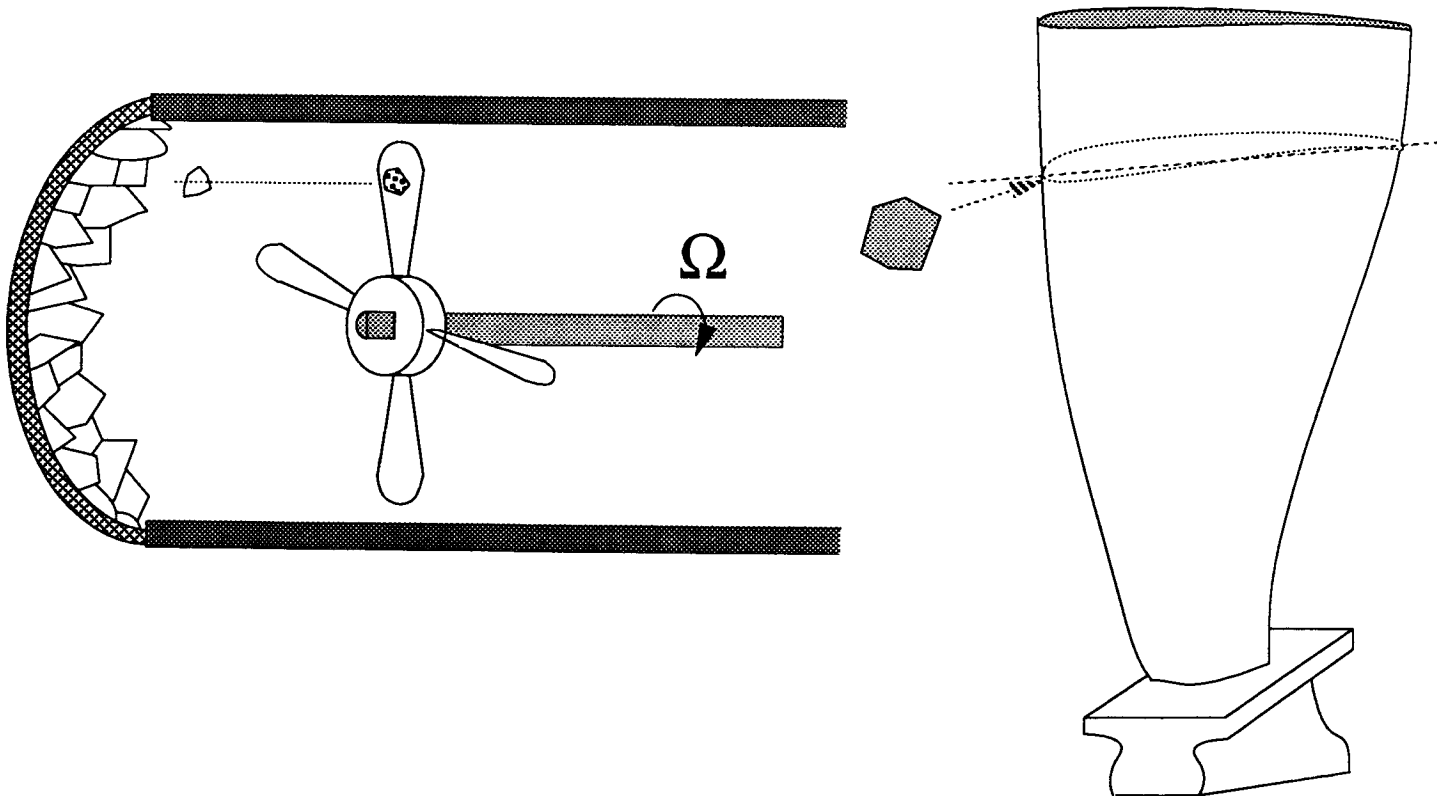


Figure 1. A Schematic of Ice Impact on an Engine Blade

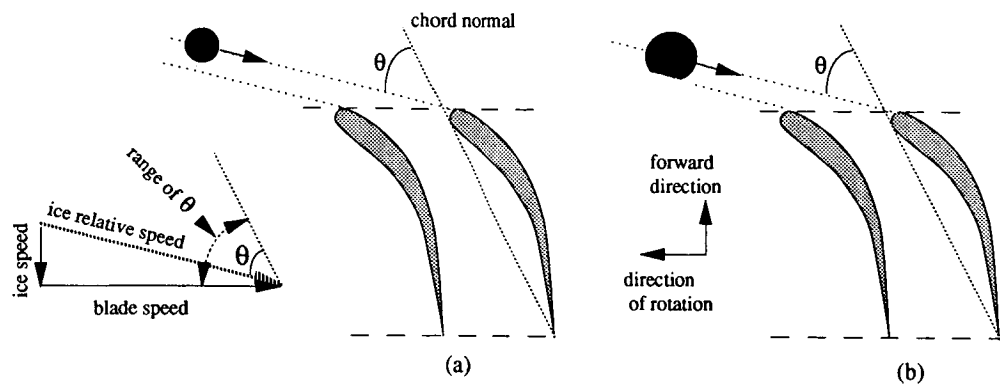


Figure 2. Geometry of Ice Impact on an Engine Blade

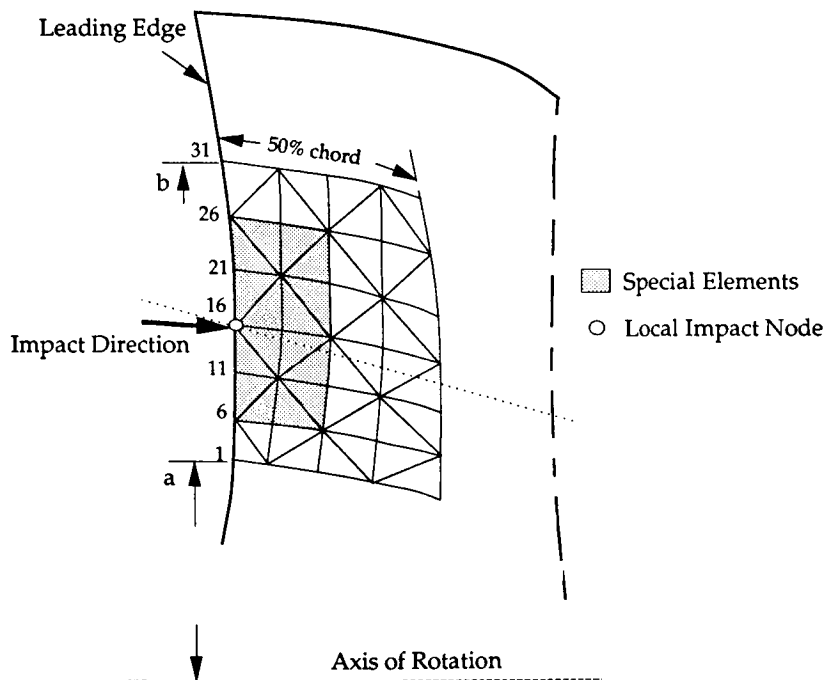


Figure 3. Finite Element Model for Local Impact Analysis

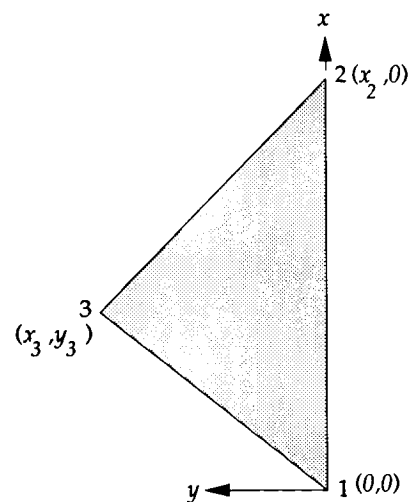


Figure 4. Finite Element Local Coordinate System

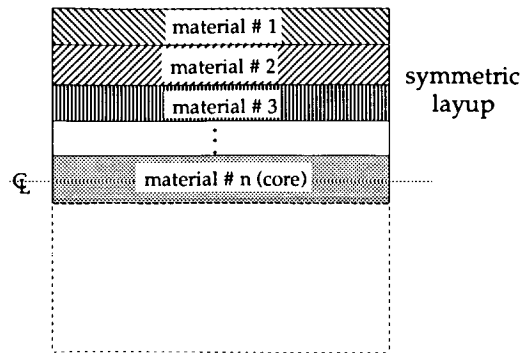


Figure 5. Symmetric Material Layer Configuration in the Composite Blade

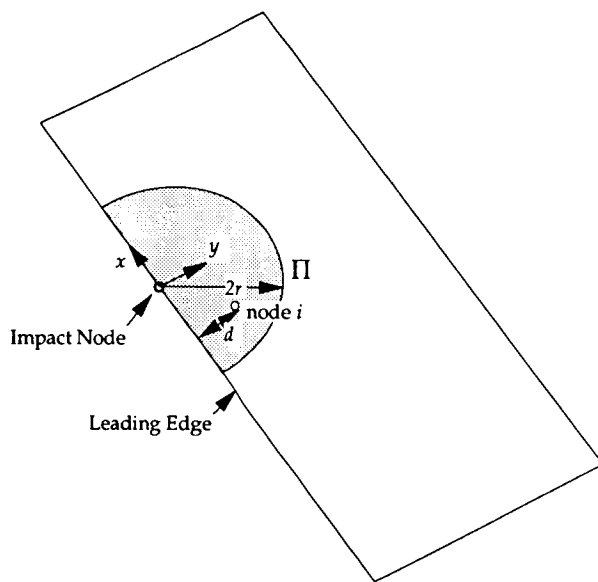


Figure 6a. Force Distribution Region in Local Impact

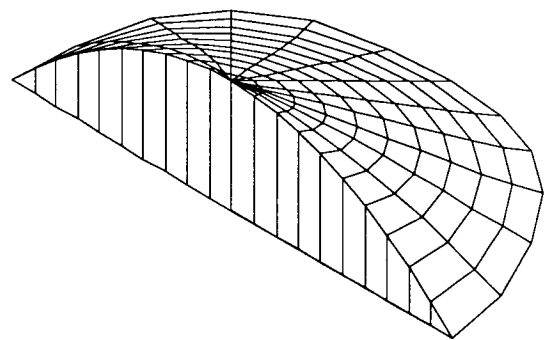


Figure 6b. Parabolic Force Distribution in the Region II

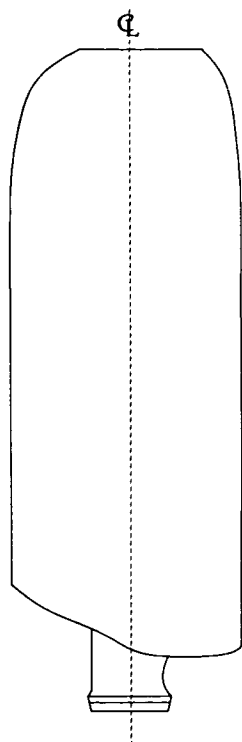


Figure 7. A Schematic of the Planform of SR-2 Model Propfan Blade

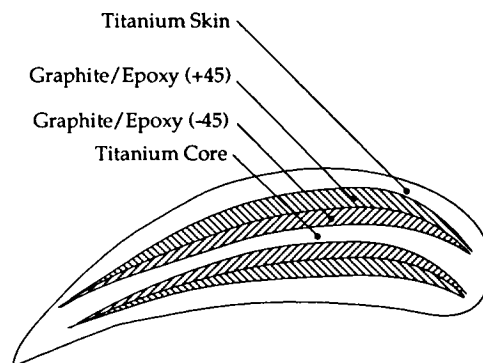


Figure 8. Material Layup of the Unswept Blade Example

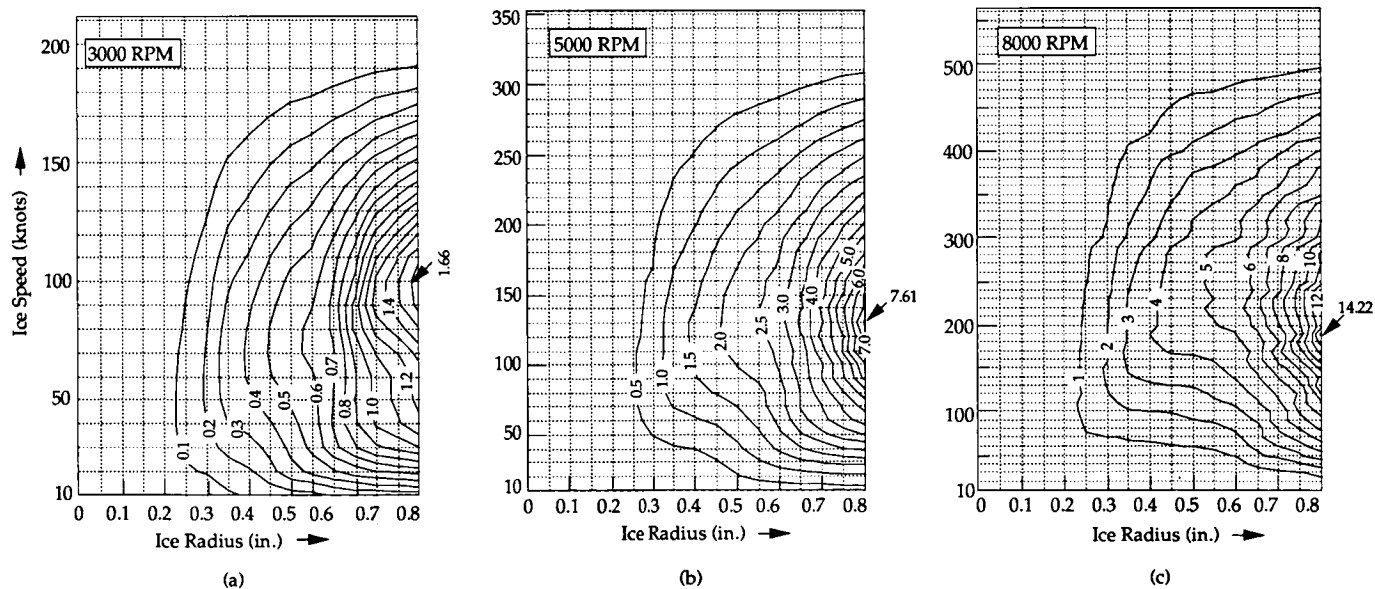


Figure 9. Average Leading Edge Strain Contours as a Function of Ice Speed and Ice Size

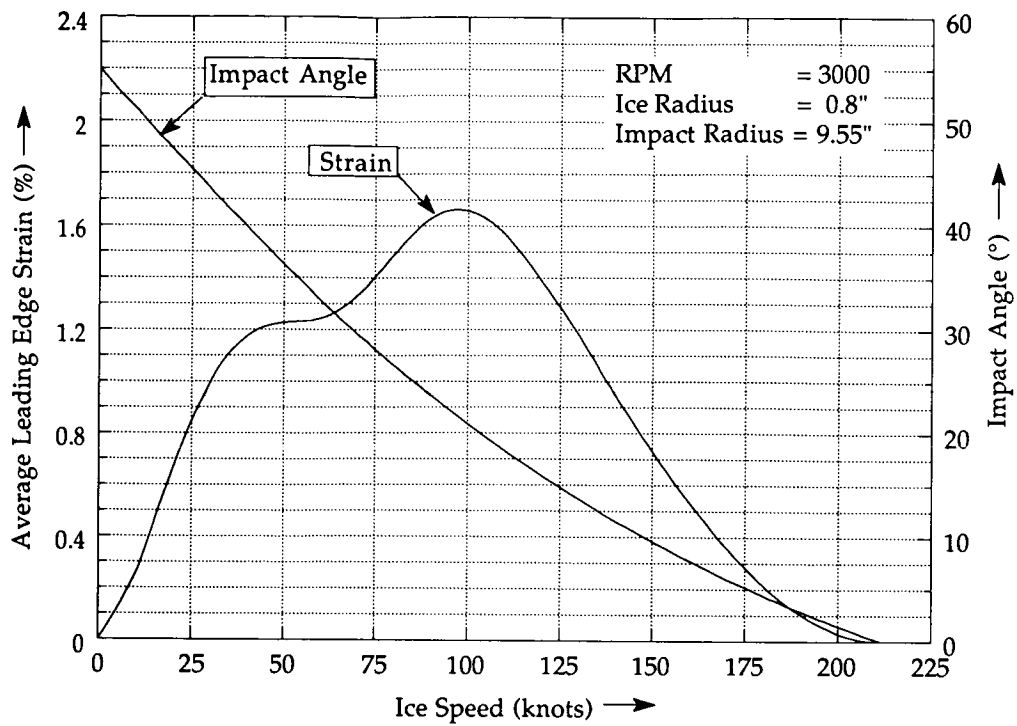


Figure 10. Variation of Average Leading Edge Strain and Impact Angle with Ice Speed

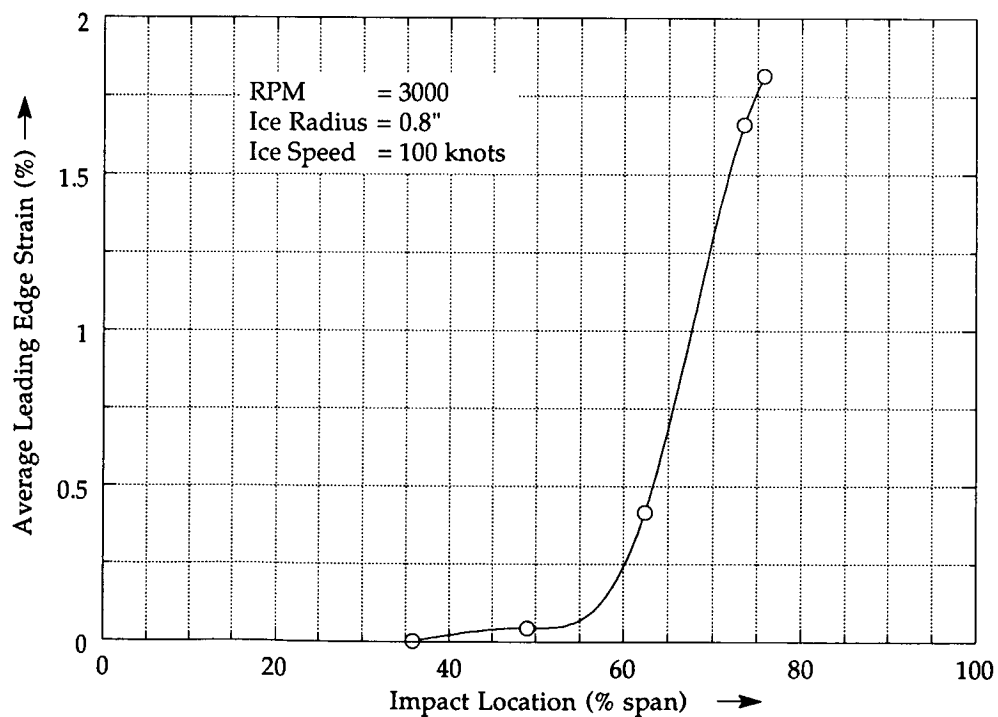


Figure 11. Effect of Impact Location on Average Leading Edge Strain

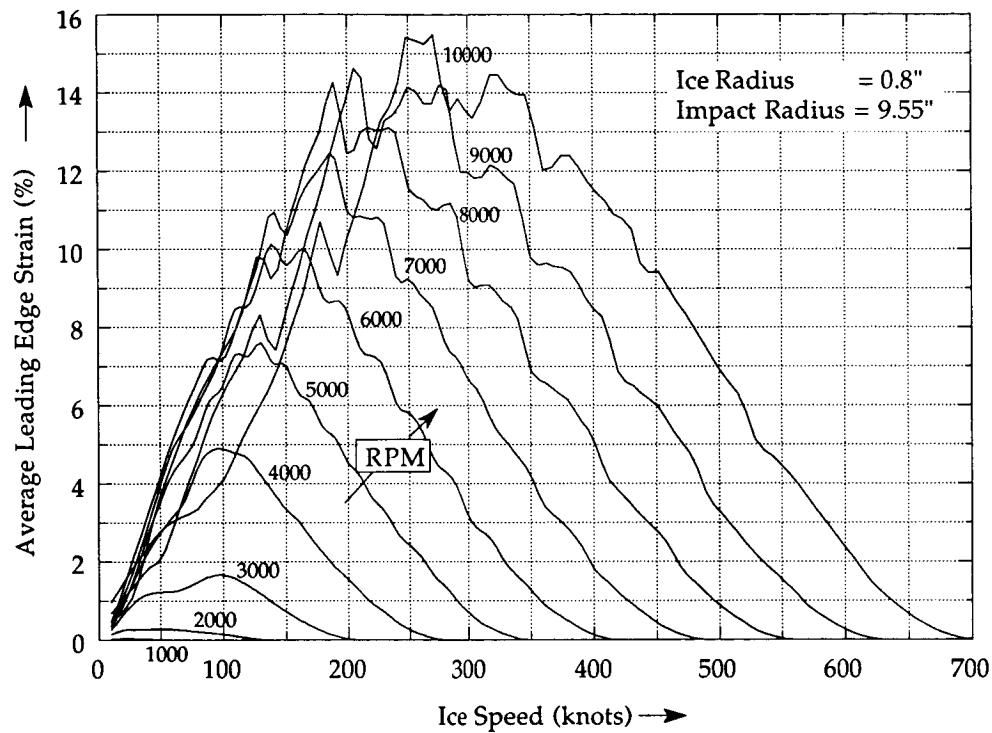


Figure 12. Effect of Ice Speed on Average Leading Edge Strain with RPM as Parameter

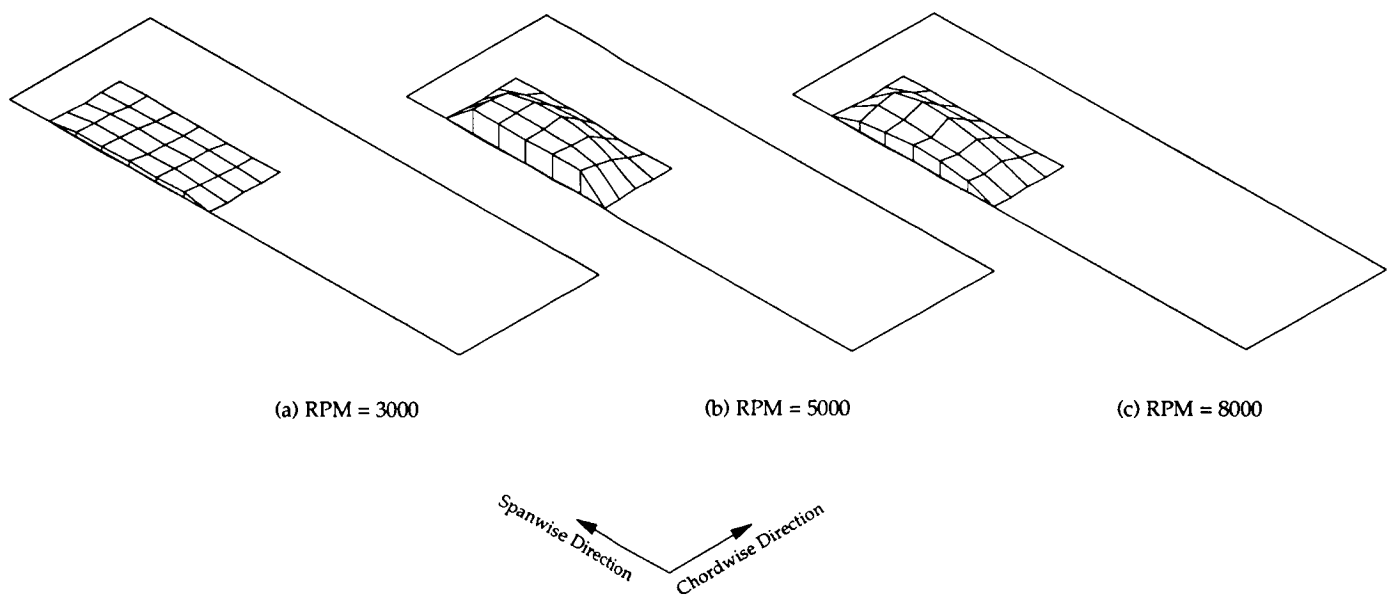


Figure 13. Absolute Out-of-Plane Displacements of Impact Region, Ice Size = 0.8", Ice Speed = 200 knots, Impact Location: 73.5% Span

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## Application of Fiber-Reinforced Bismaleimide Materials to Aircraft Nacelle Structures

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P. 15

## ABSTRACT

Existing aircraft engine nacelle structures employ advanced composite materials to reduce weight and thereby increase overall performance. Use of advanced composite materials on existing aircraft nacelle structures includes fiber-reinforced epoxy structures and has typically been limited to regions furthest away from the hot engine core. Portions of the nacelle structure that are closer to the engine require materials with a higher temperature capability. In these portions, existing nacelle structures employ aluminum sandwich construction and skin/stringer construction. The aluminum structure is composed of many detail parts and assemblies and is usually protected by some form of ablative, insulator, or metallic thermal shield.

A one-piece composite inner cowl for a new-generation engine nacelle structure has been designed using fiber-reinforced bismaleimide (BMI) materials and honeycomb core in a sandwich construction. The composite structure is 9 feet in length and 10 feet wide in cross section. The new composite design has many advantages over the existing aluminum structure. Multiple details have been integrated into the one-piece composite design, thereby significantly reducing the number of detail parts and fasteners. The use of lightweight materials and the reduction of the number of joints result in a significant weight reduction over the aluminum design; manufacturing labor and the overall number of tools required have also been reduced.

Several significant technical issues were addressed in the development of a BMI composite design. Technical evaluation of the available BMI systems led to the selection of a toughened BMI material which was resistant to microcracking under thermal cyclic loading and enhanced the damage tolerance of the structure. Technical evaluation of the degradation of BMI materials in contact with aluminum and other metals validated methods for isolation of the various materials. Graphite-reinforced BMI in contact with aluminum and some steels was found to degrade in salt spray testing. Isolation techniques such as those used for graphite-reinforced epoxy structures were shown to provide adequate protection.

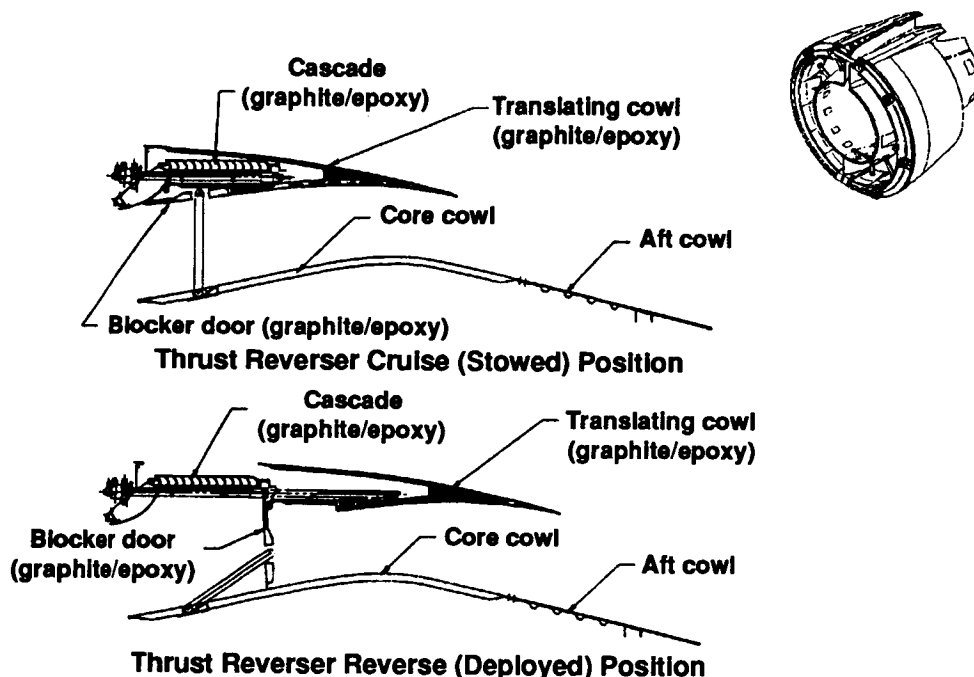
The springback and producibility of large BMI structures were evaluated by manufacturing prototype hardware which had the full-scale cross section of the one-piece composite structure. A female tooling approach was adopted to control critical aerodynamic surface tolerances. Composite tooling and laminate design were successfully combined to minimize springback.

## INTRODUCTION

Existing aircraft engine nacelle structures employ advanced composite materials to reduce weight and thereby increase overall performance. Use of advanced composite materials on existing aircraft nacelle structures includes fiber-reinforced epoxy structures and has typically been limited to regions furthest away from the hot engine core. Portions of the nacelle structure that are closer to the engine require materials with a higher temperature capability. In these portions, existing nacelle structures employ aluminum sandwich construction and aluminum skin/stringer construction. The aluminum structure is composed of many detail parts and assemblies and is usually protected by some form of ablative, insulator, or metallic thermal shield.



The usage of advanced composite materials in aircraft nacelle structures, specifically the engine thrust reverser, is illustrated in Figure 1. The thrust reverser has two concentric cowls which together form the fan duct flowpath. The outer cowl or translating cowl moves aft during landing and deploys the blocker doors which reverse the fan duct flow. The resulting reverse thrust helps to slow the aircraft. Since the translating cowl is in a region furthest away from the hot engine core, many translating cowls in service today are constructed of fiber-reinforced epoxy.



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Figure 1. Existing use of composite materials in nacelle structure

The inner cowl of the thrust reverser, however, is stationary and surrounds the hot engine core. The inner cowl provides the primary attachments of the thrust reverser to the engine, access to engine components through a series of access doors, and inlets for airflow for various engine or aircraft systems. The reverser is attached to the engine pylon at the hinge beam at the upper edges of the inner cowl. The two reverser halves are latched together at the latch beam, which is attached to the lower edges of the inner cowl. The reverser is also attached at its forward and aft ends to the engine.

Many of the inner cowls in service today consist of a core cowl, upper and lower bifurcations, and an aft cowl, as shown in Figure 2. The core cowl and bifurcations are assembled into one subassembly and the aft cowl is another. Several recent designs in service have incorporated the core cowl and bifurcations into one integral metallic structure. Various fairings and inlets are subsequently attached to the inner cowl. These assembly operations involve many detail parts and a significant number of fasteners. Therefore, during the last decade, work has focused on the use of advanced composite materials for portions of the inner cowl. Composite structures utilizing high-temperature materials such as bismaleimides and polyimides have been considered [1,2]. Development efforts on a one-piece composite inner cowl were started by Martin Marietta Aero & Naval Systems in 1985. Continuation of the effort into detail design, certification, and production of the one-piece composite inner cowl has been funded by Pratt & Whitney since 1990.

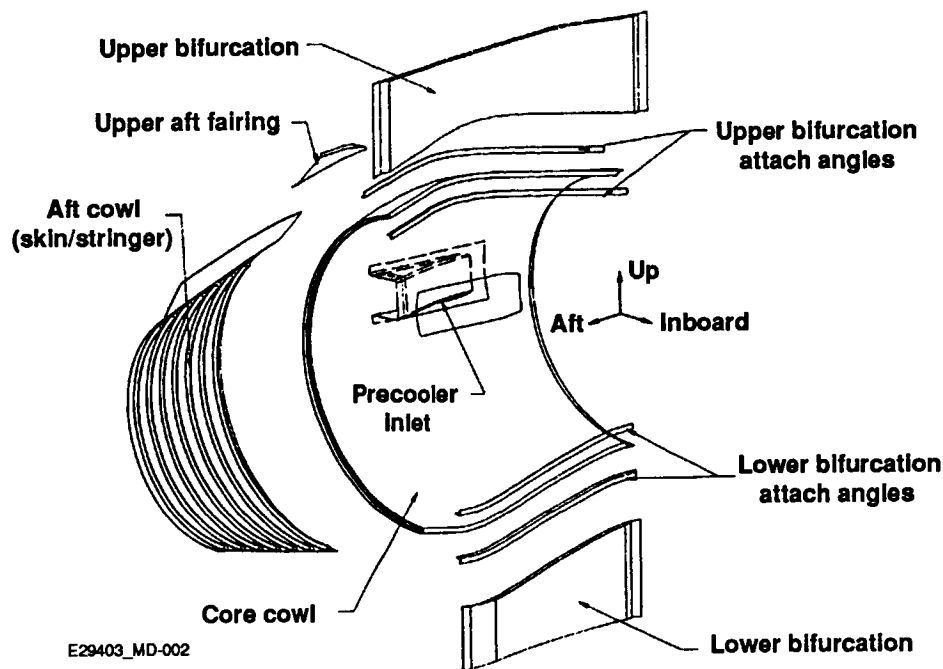


Figure 2. Present aluminum inner cowl design

The inner cowl, or fixed-structure bondment, of a new-generation engine nacelle structure features a composite sandwich construction with fiber-reinforced bismaleimide (BMI) skin materials and honeycomb core. The design employing composite materials has several advantages over the existing aluminum structure, including lower weight and fewer parts. Several significant technical issues were resolved in the development of a BMI composite design. These issues include microcracking of BMI laminates due to thermal cyclic loading, degradation of BMI laminates in contact with or in close proximity to aluminum and other metals, and springback of a large composite structure due to its cure cycle. The design, materials selection, and manufacture of the BMI one-piece inner cowl are discussed in the following sections.

## DESIGN

The one-piece composite inner cowl design, shown in Figure 3, is approximately 9 feet in length and 10 feet wide in cross section. The design integrates many of the individual components of the present inner cowls into one structure. Initial trade studies focused on integration of the core cowl, upper bifurcation, and lower bifurcation. As the trade studies proceeded, the inner cowl was extended to include the aft cowl as well as the upper aft fairing. The final design incorporated the turbine case cooling inlet as well as the precooler inlet and associated reinforcing structure.

The one-piece composite design significantly reduces the number of detail parts and the number of fasteners compared to the aluminum design. The use of lightweight materials, the ability to tailor laminate construction, and the reduction of the number of joints results in a considerable weight reduction. Trade studies show that the BMI inner cowl design is 30 to 35 pounds lighter than the aluminum design.

The one-piece inner cowl design also has manufacturing advantages over the aluminum design. The reduced number of detail parts and joints and fasteners results in reduced manufacturing labor for the inner cowl. Many manufacturing operations are consolidated into one operation by the transfer of the majority of the manufacturing to a traditionally more efficient area of production in the bonding facility. In addition, the overall number of tools required is reduced significantly.

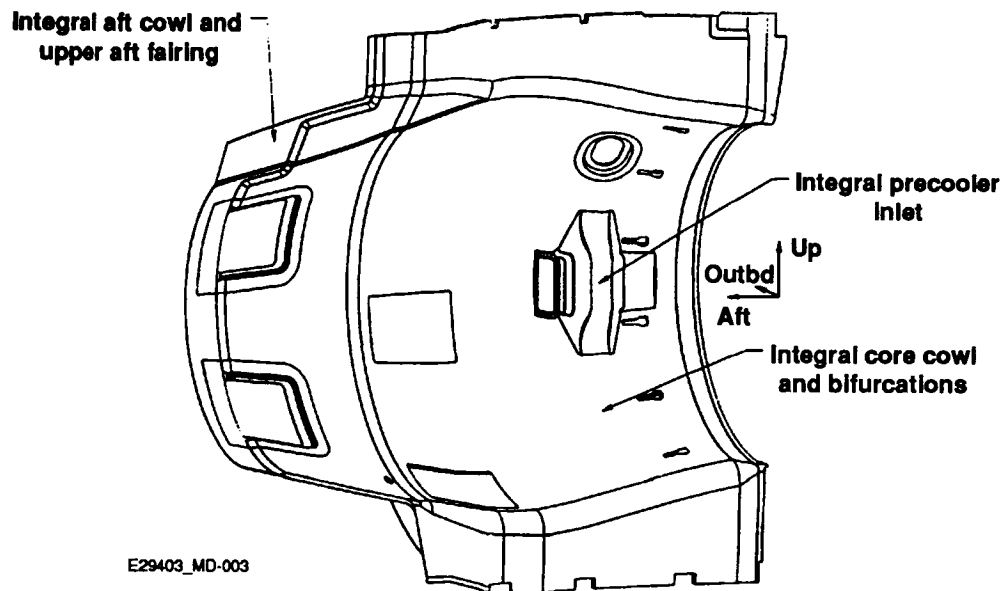


Figure 3. One-piece composite inner cowl

The longitudinal cross section of the BMI inner cowl is shown in Figure 4. The inner cowl consists of a precured outer skin, local precured reinforcing doublers, honeycomb core, and a precured inner skin. Because the forward section of the inner cowl is part of the sound-suppression system for the engine, this portion has a perforated BMI outer skin. Since the aft portion of the inner cowl is beyond the fan duct where no sound suppression is required, this portion has a solid BMI outer skin. Local BMI doublers are employed to reinforce cutouts and attachments. These doublers fit into recessed areas of the core.

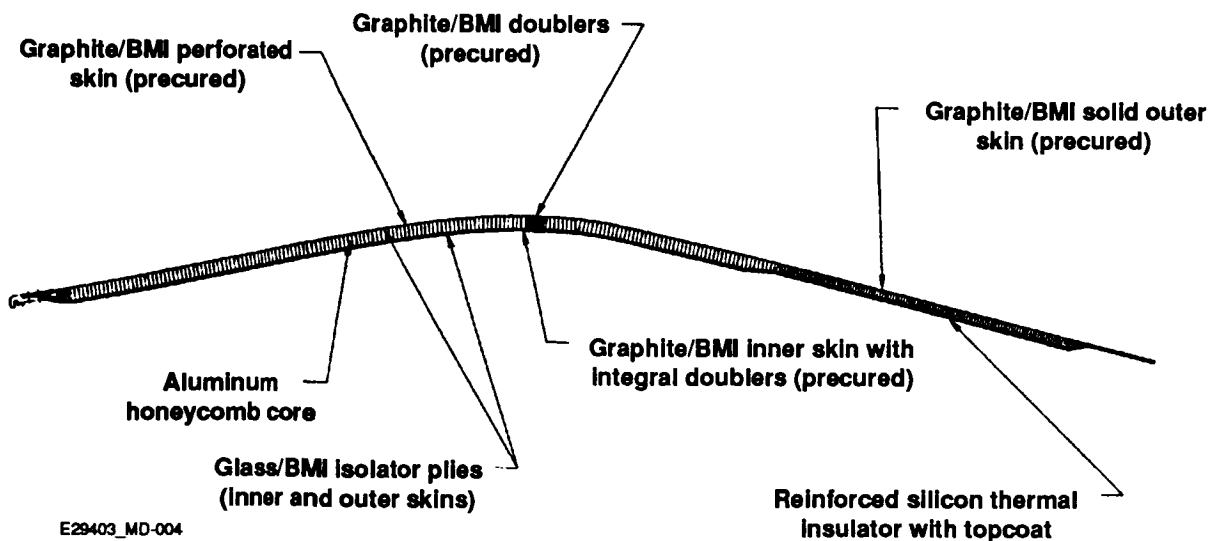


Figure 4. Longitudinal cross section of one-piece composite inner cowl

The solid inner skin is continuous over the full length of the inner cowl and incorporates edge closeouts. The skin is tailored to minimize weight while meeting the load requirements. Hence, local integral doublers are precured as part of the inner skin. These doublers are placed on the inner surface of the inner skin, thereby eliminating the need for recesses for doublers in the core. As a result, the core has a smooth surface for improved fitup of the core to the skin. Finally, the inner cowl is protected by a reinforced silicon thermal insulator sprayed onto its inner surface.

A typical circumferential cross section is shown in Figure 5. The inner cowl is continuous from the upper bifurcation to the lower bifurcation. At the bifurcation-to-core cowl joint is a 3-inch radius with local doublers for reinforcement. The integral precooler inlet, also shown in Figure 5, provides an effective circumferential load path across the large precooler cutout, which for previous designs has required significant reinforcing structure. Both the core cowl-to-bifurcation joint and the integral precooler inlet result in weight reduction.

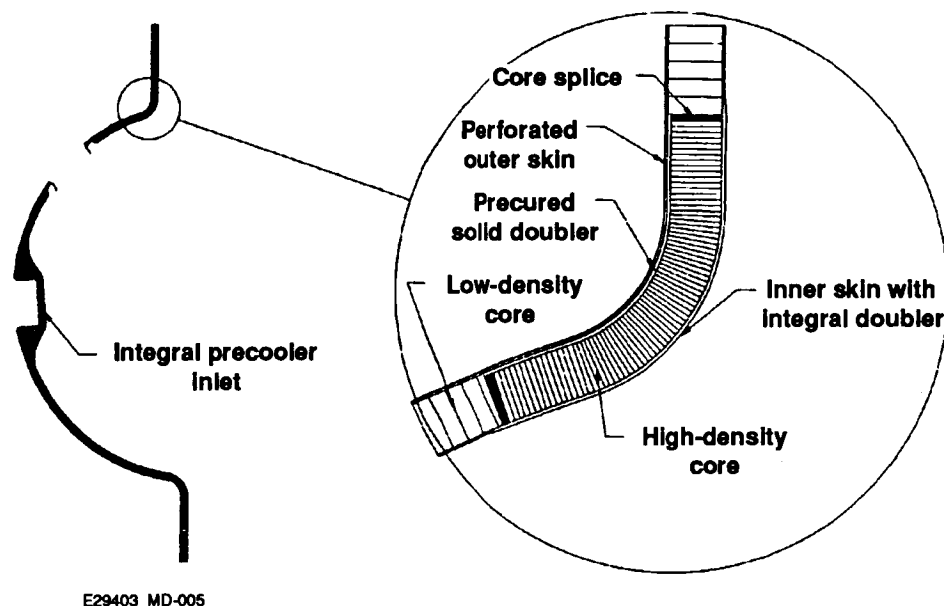


Figure 5. Circumferential cross section of one-piece composite inner cowl

Controlling the temperature of the inner skin below the design limit of 350° F proved to be a key technical issue and was highly dependent on the choice of honeycomb core material. Various honeycomb core materials were considered, including glass/polyimide, glass/phenolic, and aluminum. Figure 6 shows the temperature profile versus engine station for the aluminum and glass/polyimide cores. The glass-reinforced core materials have low thermal conductivity, resulting in inner skin temperatures above the design limits. Significant weight penalties are projected for the glass-reinforced core design due to the need for increased thermal protection. The aluminum core, which has the best conductivity, results in the lowest inner skin temperatures. Graphite-reinforced polyimide core resulted in inner skin temperatures slightly higher than those for the aluminum core design. A comparison of the cost of the various core options showed that the glass and graphite-reinforced core options were considerably more expensive than the aluminum core design.

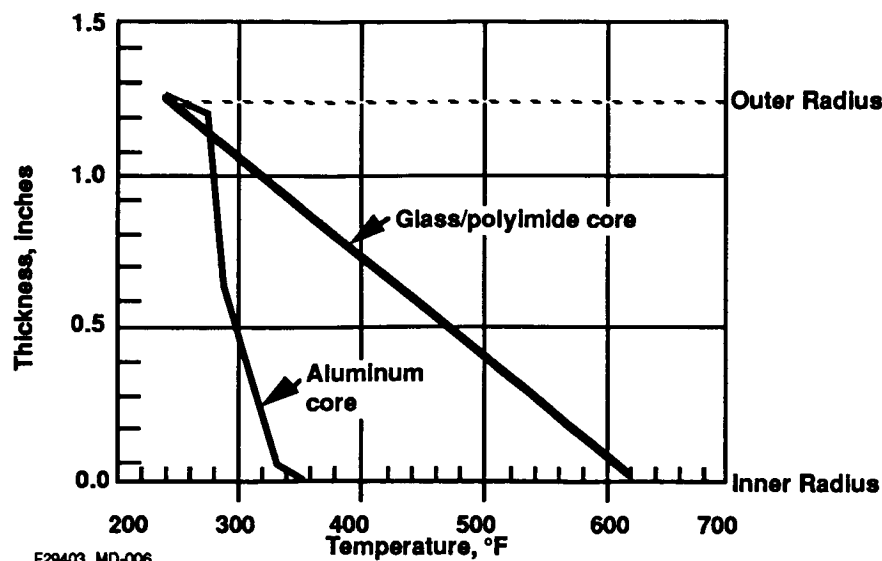
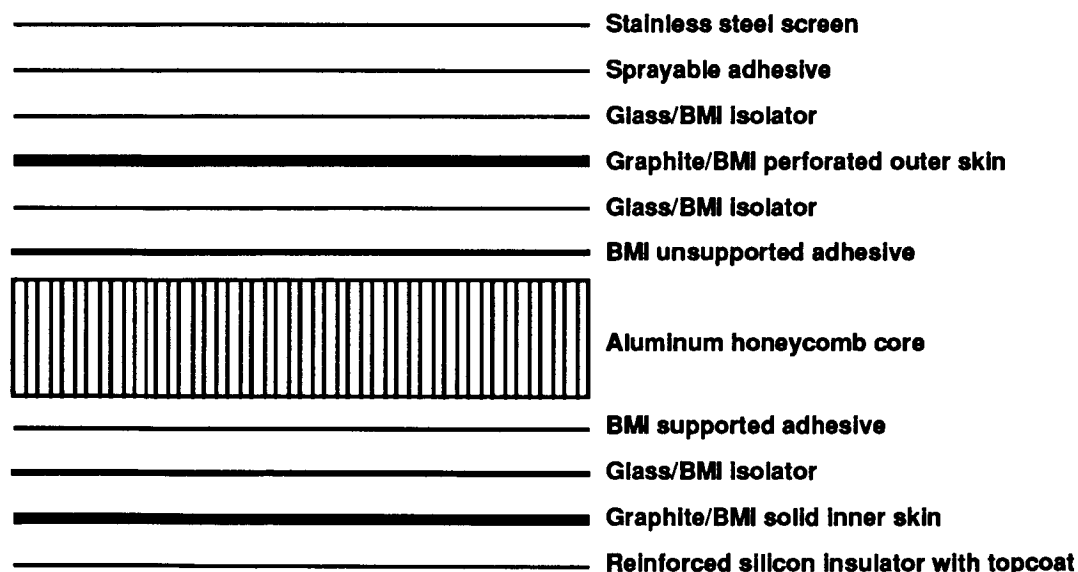


Figure 6. Effect of core material on temperature distribution

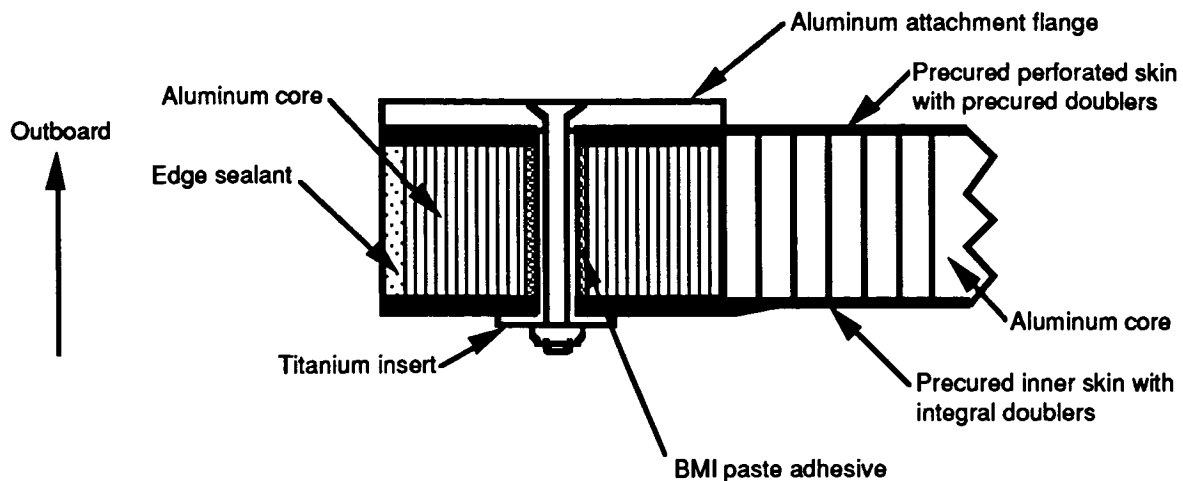
Corrosion protection of the inner cowl materials is another key technical issue. Moisture ingress is possible in some areas of the inner cowl due to the perforated outer skin. Both corrosion of the aluminum core and degradation of the graphite/BMI laminates were addressed. Figure 7 shows the typical isolation techniques for the inner cowl. Aluminum core is isolated from both the inner and outer graphite/BMI skins by a glass/BMI isolation layer. A glass/BMI layer is also placed on the outer surface of the perforated skin to isolate it from the stainless steel screen required for sound suppression. In addition, the aluminum core is treated with a protective coating to enhance its corrosion resistance.



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Figure 7. Isolation of aluminum core and steel screen

Attachments to the inner cowl are also designed with corrosion protection in mind. A typical attachment to the inner cowl is shown in Figure 8. Glass/BMI layers are placed locally on the BMI skins to isolate them from aluminum attachment hardware, and titanium inserts are used to further prevent BMI degradation. Titanium fasteners are used for attachments in areas of the inner cowl which consist only of laminate.



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Figure 8. Insert design for composite inner cowl

## MATERIALS

Several technical material issues had to be addressed to incorporate the fiber-reinforced BMI materials in conjunction with aluminum core into the composite inner cowl design. Major issues, as previously stated, included BMI microcracking, BMI degradation, and aluminum core corrosion.

### BMI Microcracking

First-generation BMI systems, toughened BMI systems, and high-temperature epoxies were evaluated for laminate mechanical properties after thermal cycling and for laminate microcracking as a function of thermal cycling and isothermal aging. A thermal cyclic profile, shown in Figure 9, was developed to represent the thermal loading of the inner cowl. It includes cycles from  $-67^{\circ}\text{F}$  to  $375^{\circ}\text{F}$  and from  $11^{\circ}\text{F}$  to  $375^{\circ}\text{F}$  with heatup rates of 20 degrees per minute, typical of the engine thrust reverser. The laminate used was a mix of fabric (f) and unidirectional tape (t),  $[90_f, 90_t, 45_f, -45_f, 90_t, 90_f, 90_t, -45_f, 45_f, 90_t, 90_f]$ . Panels with this laminate construction were fabricated and inspected. Specimens were prepared and exposed up to 1000 thermal cycles. The first-generation BMI and the epoxy systems exhibited an increase of microcracking as a function of thermal cycles while the toughened BMI systems essentially showed no evidence of microcracking.

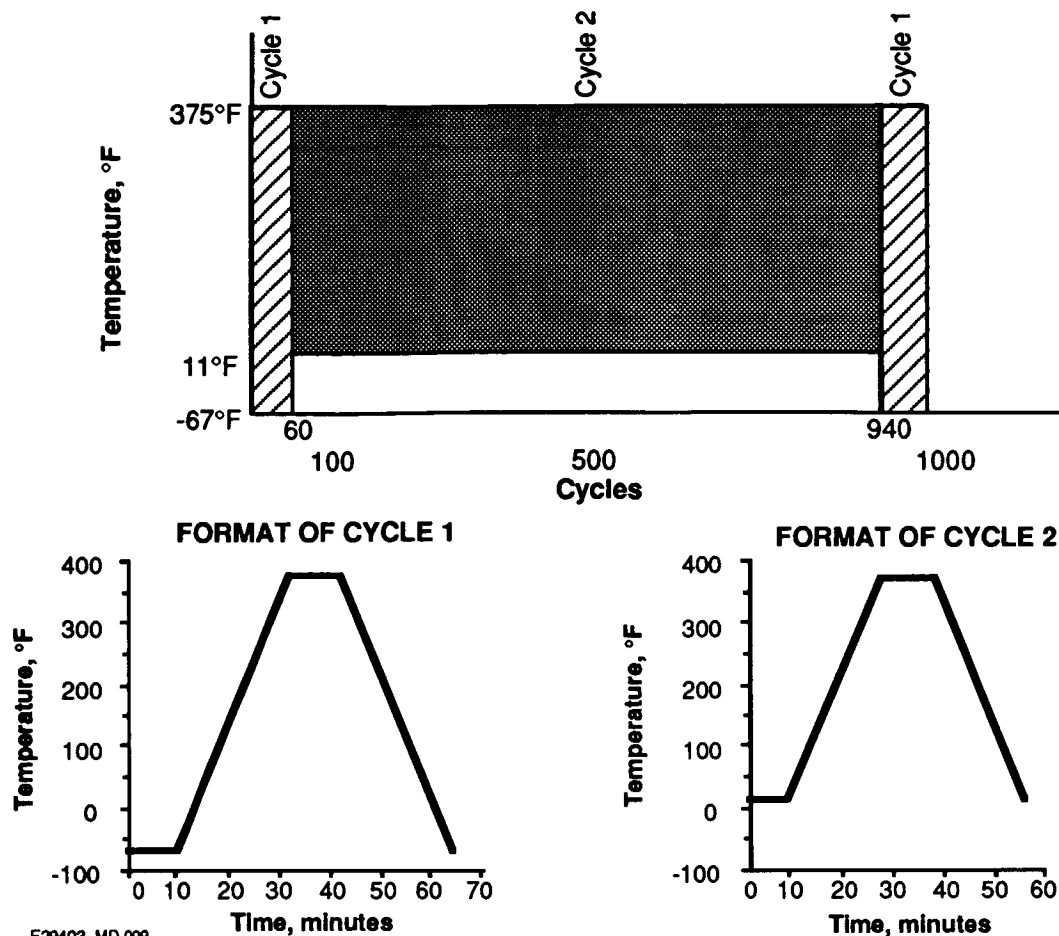
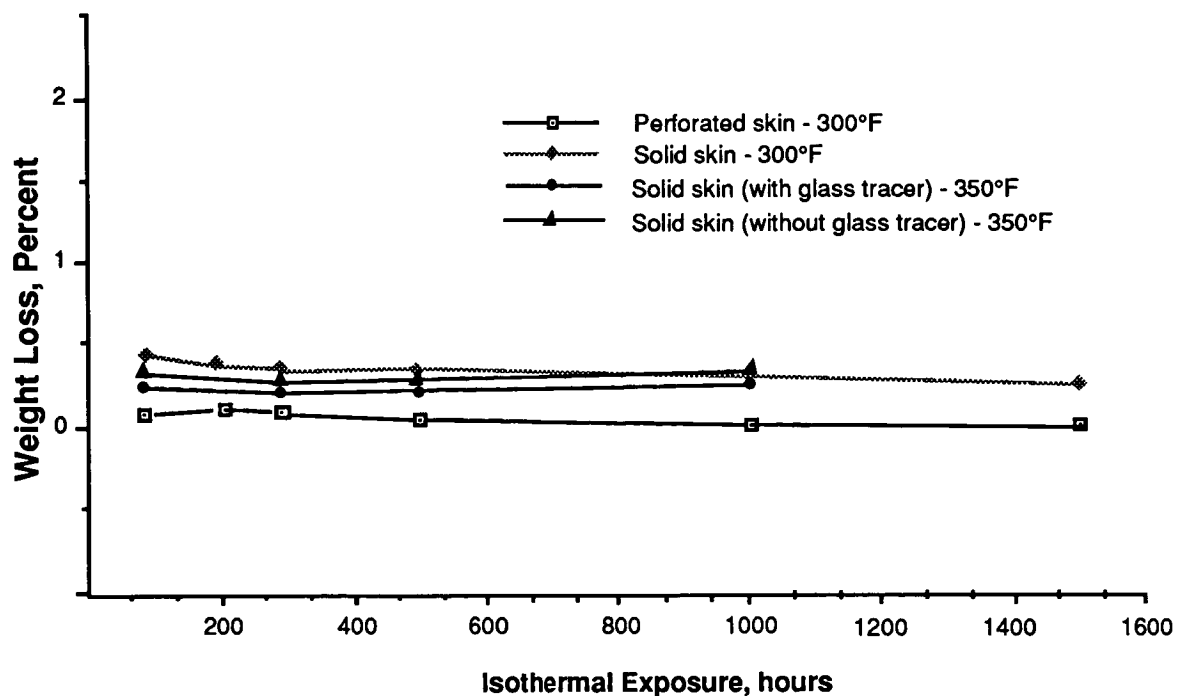


Figure 9. Thermal cycle profile for microcracking evaluation

In addition to the visual examination for microcracks, the residual compressive and interlaminar shear strength of the laminate after thermal cycling was measured. The specimens were tested at 350° F after 240 hours exposure at 160° F and 97 percent relative humidity. Compressive testing was performed using an IITRI (Illinois Institute of Technology Research Institute) fixture in accordance with the ASTM for Compressive Properties of Unidirectional or Cross-Ply Fiber-Resin Composites (D3410-82). Interlaminar shear testing was performed in accordance with the ASTM for Apparent Interlaminar Shear Strength of Parallel Fiber Composites by Short Beam Methods (D2344-84). Test results showed approximately a 10 percent drop in strength after 1000 thermal cycles. Isothermal aging of specimens with the laminate construction discussed above was also performed. Specimens exposed to 375° F for 800 hours showed no evidence of microcracking.

In addition, the performance of the perforated BMI outer skin was evaluated. Testing focused on the perforated skin because of the potential for the increased surface area to cause increased microcracking. Isothermal aging tests were performed to study the effect of temperature and the effect of perforations on the BMI outer skin; however, the design temperature for the perforated outer skin was only 300° F. Solid and perforated laminate specimens with a  $[(90)_4]$  laminate construction, the downselected configuration, were fabricated and inspected and isothermally aged at 300° F. Solid laminates were also isothermally aged at 350° F. The results of weight loss measurements are shown in Figure 10. Weight loss was less than 1 percent after 1500 hours at 300° F for both the solid and perforated laminates.



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Figure 10. Weight loss results for BMI microcracking evaluation

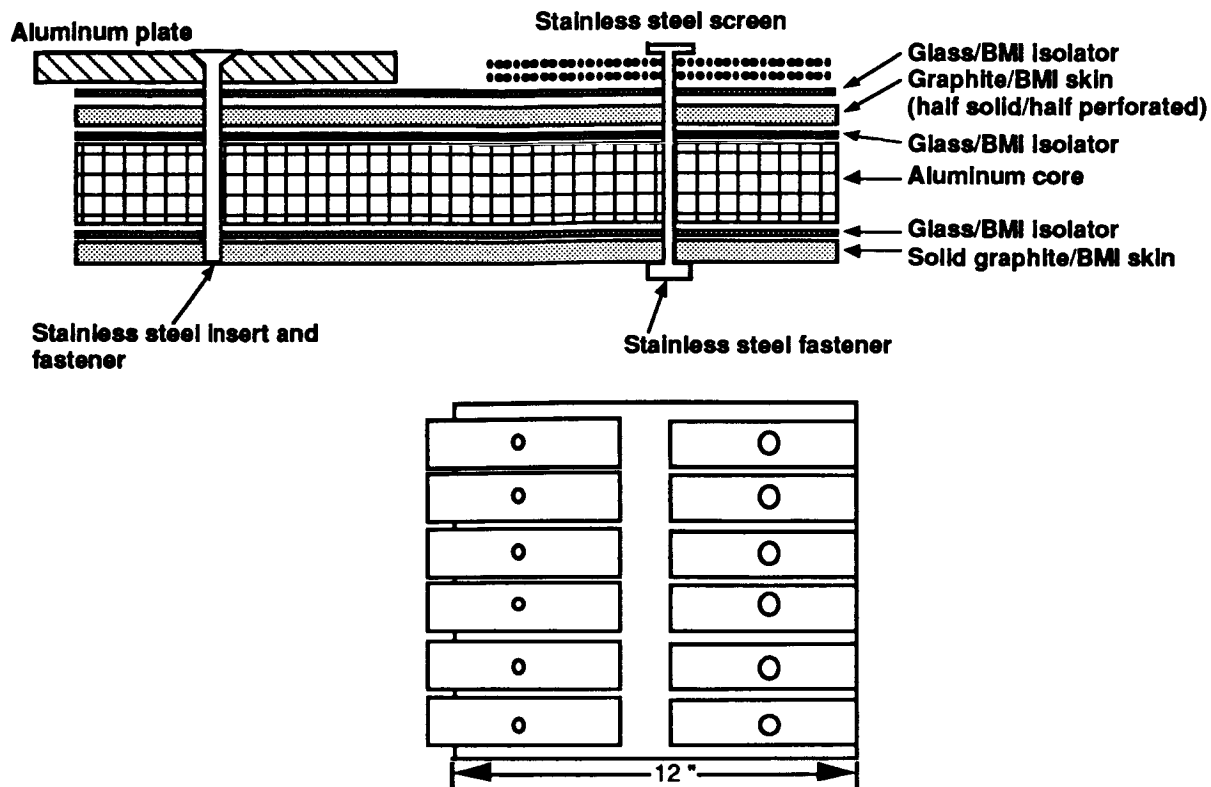
A toughened BMI material was selected for the composite inner cowl based on these results. Besides its higher resistance to microcracking under thermal cyclic loading and isothermal aging, the toughened BMI material produces an improved damage-tolerant structure.

### BMI Degradation

Recent studies have shown that graphite/BMI laminates can degrade when in contact with aluminum and various other metals under certain conditions. The degradation is caused by hydrolysis of imides by hydroxide ions [3]. The phenomenon will not be described further; rather, it is evaluated and suitable methods to prevent it are presented. This effort was performed in two parts. The first part was performed by exposing laminates to an electrolyte solution (salt water) while in contact with various other materials. Graphite-reinforced BMI laminates degraded when coupled with aluminum and some steels, but no degradation was observed in laminates in contact with titanium and glass-reinforced BMI materials.

The second part of the evaluation was performed using panels representative of the inner cowl and its attachments. These sandwich construction panels, shown in Figure 11, were salt spray tested in accordance with the ASTM for Salt Spray (Fog) Testing (B117-85). These sandwich construction panels had perforated BMI outer skins and solid BMI inner skins. Aluminum core was evaluated with and without glass/BMI isolation layers. All panels had a steel screen bonded to the perforated skin and an aluminum plate attachment. However, none of the panels had the insert design which is incorporated in the final design. Instead, fasteners were installed directly through the core, representing a conservative test scenario.





| Test Panel Type | Hinge Plate Type - Isolator |     | Fastener Installation | Core Isolator | Screen Isolator | BMI Type         |
|-----------------|-----------------------------|-----|-----------------------|---------------|-----------------|------------------|
| I               | Bare                        | No  | Dry                   | No            | No              | First Generation |
| II              | Anodized                    | Yes | Wet                   | Yes           | Yes             | First Generation |
| III             | Anodized                    | Yes | Wet                   | Yes           | Yes             | Toughened        |

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Figure 11. BMI degradation sandwich panel configuration

The results of the salt spray tests up to 2000 hours are shown in Figure 12. The BMI laminates in the panels without isolation degraded. In contrast, results for the isolated panels showed that isolation techniques such as those used for graphite-reinforced epoxy structures protect adequately against BMI degradation.

| TEST PANEL TYPE                 | EXPOSURE TIME                                                                        |                                                                                    |                                                                             |                                                                               |
|---------------------------------|--------------------------------------------------------------------------------------|------------------------------------------------------------------------------------|-----------------------------------------------------------------------------|-------------------------------------------------------------------------------|
|                                 | 150 HOURS, NOMINAL                                                                   | 500 HOURS, NOMINAL                                                                 | 1000 HOURS, NOMINAL                                                         | 2000 HOURS, NOMINAL                                                           |
| WITHOUT ISOLATOR                | MODERATE DEGRADATION BETWEEN ALUMINUM AND STAINLESS STEEL AND ON BACK OF BOTTOM SKIN | SEVERE DEGRADATION BETWEEN ALUMINUM AND STAINLESS STEEL AND ON BACK OF BOTTOM SKIN | SEVERE DEGRADATION ON ALL EXPOSED SURFACES                                  | SEVERE DEGRADATION ON ALL EXPOSED SURFACES                                    |
| WITH ISOLATOR                   | NO BMI DEGRADATION                                                                   | NO BMI DEGRADATION                                                                 | NO DEGRADATION ON PROTECTED SKIN, SLIGHT ON UNPROTECTED BACK OF BOTTOM SKIN | NO DEGRADATION ON PROTECTED SKIN, MODERATE ON UNPROTECTED BACK OF BOTTOM SKIN |
| WITH ISOLATOR AND TOUGHENED BMI | NO BMI DEGRADATION                                                                   | NO BMI DEGRADATION                                                                 | NO BMI DEGRADATION                                                          | NO DEGRADATION ON PROTECTED SKIN, MODERATE ON UNPROTECTED BACK OF BOTTOM SKIN |

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Figure 12. Graphite/BMI degradation tests results

#### Aluminum Core Corrosion

Methods to enhance the corrosion resistance of aluminum core were investigated. As discussed previously, aluminum core was selected for the composite inner cowl due to its favorable thermal properties, in addition to weight and cost considerations. Aluminum core has been successfully used in the aircraft industry with graphite/epoxy solid skins, protected by glass/epoxy isolators and corrosion-inhibiting sealants and coatings. However, the case is different for the composite inner cowl since the perforated outer skin allows moisture ingress into the structure. The structure is also different due to the BMI skins and the required post cure for the BMI adhesives.

Aluminum cores with various protective coatings were considered in an effort to enhance corrosion resistance for this application. Both aluminum cores dipped in chromated epoxy primers and aluminum cores with phosphoric acid anodize were evaluated. In addition, both types of aluminum cores were also coated with a second layer of epoxy primer. This secondary layer was considered a sacrificial layer to improve corrosion resistance.

Testing was performed to evaluate the effects of the post cure on the aluminum core and the various protective coatings. Specimens of aluminum core with various protective coatings were prepared and exposed to post cures ranging from none to up to 450° F. Core shear tests were performed in accordance with the ASTM for Shear Test in the Flatwise Plane of Flat Sandwich Constructions or Sandwich Cores (C273-80). Salt spray tests were performed in accordance with the ASTM for Salt Spray (Fog) Testing (B117-85). Results showed that the higher post cure temperatures caused a reduction in mechanical properties and adversely affected the protective coatings. An optimum post-cure temperature less than 450° F was selected and any strength reductions were reflected in the design allowables. The aluminum cores with a secondary epoxy primer layer performed the best, exhibiting no coating degradation after 1000 hours of salt spray exposure.

Salt spray testing of sandwich panels with aluminum core and the various protective coatings is currently in process. These sandwich panels include the isolation techniques incorporated into the composite inner cowl

design and the optimized process parameters described above. Results of these salt spray tests will provide validation of the design and the optimized processing parameters.

## MANUFACTURING

A major manufacturing concern was the springback of the large BMI structure. Minimization of springback of the one-piece inner cowl would assure dimensional control of the contour of the external flow surface. In addition, dimensional control would facilitate the assembly of other hardware to the inner cowl. Springback and producibility of the one-piece inner cowl were evaluated by manufacturing prototype hardware. The prototype selected was a 3-foot-long segment with the full-size cross section and contour of the inner cowl.

Two prototypes were fabricated early in the design process using both male and female tooling approaches to evaluate the effects of tooling on springback. The first prototype was fabricated utilizing a male composite tooling approach. A master model was manufactured and a graphite/epoxy splash was constructed; the splash was then used to fabricate the graphite/epoxy tooling. Photogrammetric inspection was successfully employed to verify the contour of the tool.

Precured inner and outer skins and precured doublers were fabricated and inspected. The precured details were assembled on the male tool along with a coremat into a final assembly. The assembly was cured at autoclave temperature and pressure and then postcured freestanding in an oven. Springback was measured using photogrammetric inspection and found to be larger than desired.

The second prototype was fabricated utilizing a female composite tooling approach, which allowed more control of the flow surface, the outer surface of the inner cowl. Composite female tools were manufactured and inspected. The design of the prototype was revised to incorporate lessons learned from the first prototype. The local reinforcement at attachments was redesigned to minimize the warping effect of unidirectional tape plies. All doublers of the inner skin were moved to the inner surface, so that coremat fitup with doublers was now required only on the outer surface of the coremat. The final assembly, shown in Figure 13, was fabricated, cured at autoclave temperature and pressure, and post cured freestanding in an oven. Springback, measured by photogrammetric inspection, was found to be acceptable.

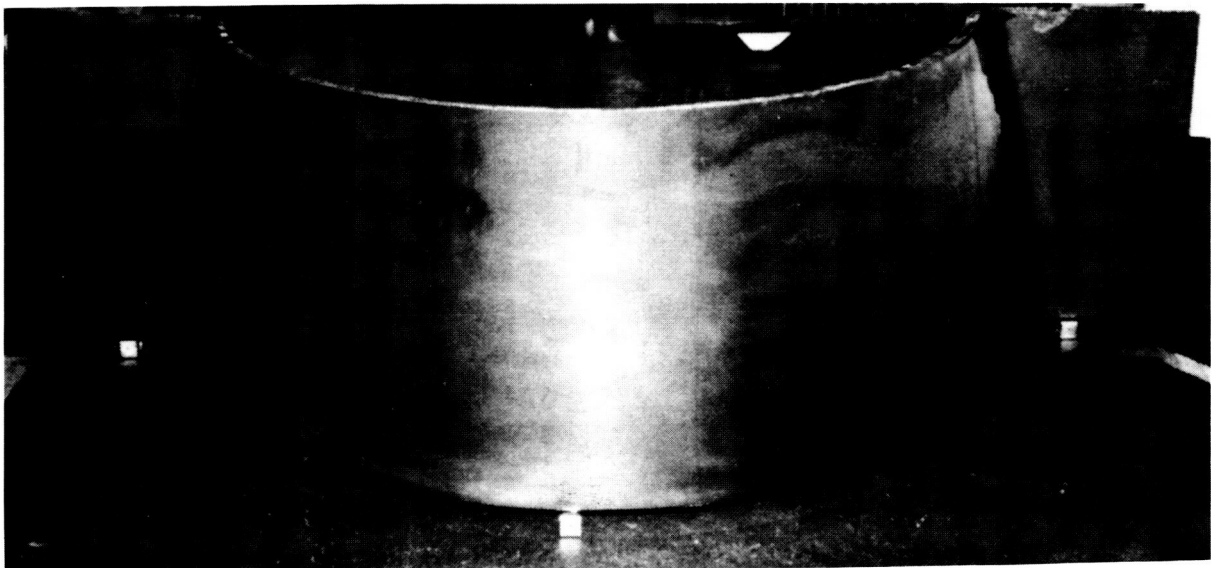


Figure 13. Prototype BMI inner cowl hardware.

Based upon the results with the two prototypes, a female tooling approach was adopted for the one-piece inner cowl. The definition of flow surface contour, prepared using CATIA, was used to prepare full-scale master models for left- and right-hand inner and outer skin tools. Splashes fabricated on these master models using graphite/epoxy materials were utilized to fabricate the graphite/epoxy tools shown in Figure 14. Photogrammetric inspection of models, splashes, and tools was successfully performed to verify flow surface contours.

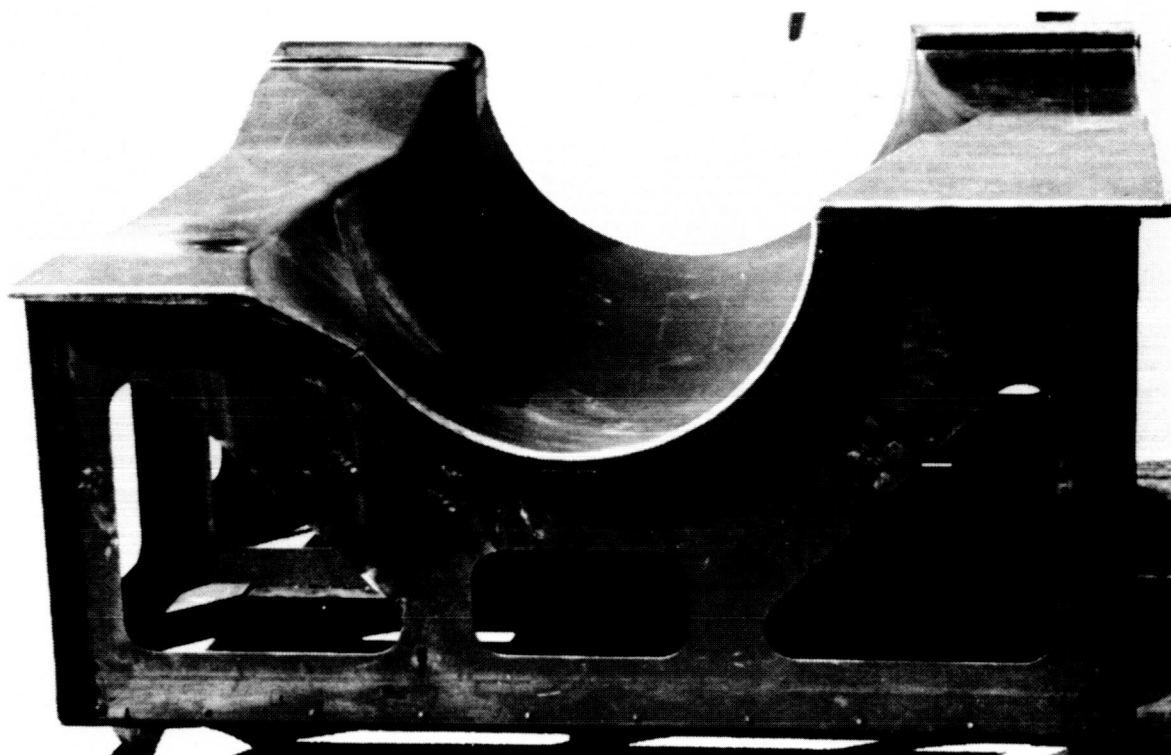


Figure 14. Full-scale composite tooling for inner cowl.

During the prototyping effort, Martin Marietta Aero & Naval Systems developed a proprietary process for the fabrication of large perforated BMI skins. Conflicting acoustic and structural requirements had to be simultaneously satisfied since the outer skin had to be perforated to meet the acoustic requirements and also had to have adequate compressive strength at elevated temperature to perform as a load-carrying member of the sandwich construction. The process for fabrication of the perforated BMI skins provides uniform hole diameter and spacing, minimizes the reduction in compressive strength due to the perforations, and can be applied to complex contoured geometries.

## SUMMARY AND CONCLUSIONS

A one-piece composite inner cowl for new-generation engine nacelle thrust reversers has been designed using fiber-reinforced bismaleimide (BMI) materials and aluminum honeycomb core in a sandwich construction. The design employing composite materials has many advantages over the existing aluminum structure. Multiple details have been integrated into the composite design, thereby significantly reducing the number of detail parts and the number of fasteners. The use of lightweight materials and the reduction of the number of joints have resulted in a significant weight reduction over the aluminum design. Trade studies show a reduction in weight of 30 to 35 pounds per reverser for the BMI design over the aluminum

design. The manufacturing labor and the overall number of tools required have also been reduced. Many manufacturing operations are consolidated into one operation by the transfer of the majority of the manufacturing to a traditionally more efficient area of production in the bonding facility.

Many significant technical and manufacturing issues were identified and resolved, including microcracking of BMI laminates due to thermal cyclic loading, degradation of BMI laminates in contact or in close proximity to aluminum and other metals, and springback of a large composite structure due to its cure cycle. Technical evaluation of microcracking of BMI laminates showed no cracking of the toughened BMI material after 1000 thermal cycles and after 800 hours of isothermal exposure at 375° F. Testing of perforated BMI laminates showed no microcracking after 1500 hours of isothermal aging at 300° F.

Technical evaluation of the degradation of BMI materials in contact with aluminum and other metals validated methods for isolation of the various materials. Graphite-reinforced BMI in contact with aluminum and some steels was found to degrade in salt spray testing. Glass-reinforced BMI materials and titanium did not cause the degradation. Isolation techniques such as those used for graphite-reinforced epoxy structures were shown to provide adequate protection.

Minimization of springback of the one-piece inner cowl was achieved through composite tooling and laminate design modifications and was demonstrated on prototype hardware. A proprietary process to manufacture BMI perforated skins was developed and validated by full-scale hardware fabrication.

Testing towards completion of certification is continuing. Tests for BMI microcracking, BMI degradation, and aluminum core corrosion are being continued to additional cycles and/or hours to obtain additional engineering information. Core corrosion tests are also focused on the effect of local impact damage and the establishment of maximum operating time before repair. The one-piece composite inner cowl is presently being manufactured and will be certified in 1993.

## ACKNOWLEDGMENTS

The authors wish to thank all members of the Fixed Structure Bondment Integrated Product Team, one of several concurrent engineering teams of the Pratt & Whitney 4168 Thrust Reverser Program at Martin Marietta Aero & Naval Systems. Special thanks are extended to Robert Kline and Lawrence Kline for their efforts in design and manufacturing, respectively. Special thanks are also extended to Mark Alberding and Rick Sirian for their efforts in BMI microcracking and aluminum core corrosion, respectively. The authors thank Matt Patterson and all members of the Pratt & Whitney staff for their support and guidance throughout the program. The authors also wish to thank the technical staffs of the many material suppliers for their significant efforts and assistance in material screening and evaluation.

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**Innovative Design and Fabrication of  
Composite Canopy Frames**

Robert L. Chu  
Michael C. Y. Niu  
Lockheed Aeronautical Systems Company



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ASSEMBLY INDUCED DELAMINATIONS IN COMPOSITE STRUCTURES<sup>1</sup>

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## SUMMARY

Experimental and analytical studies of the development of delaminations around fastener holes in composite structures are presented. This type of delamination is known to occur in composite skins that are mechanically fastened to poorly mating substructure. Results of an experimental study to determine the resistance of laminates to the initiation of assembly induced delaminations and the residual strength of assembly damaged coupons are presented for AS4/3501-6, IM7/8551-7A, and AS4/PEEK material systems.

A survey of existing analytical models for predicting the residual strength and stability of delaminations is presented, and the development of a new model for predicting the initiation of delaminations around a fastener hole is outlined. The fastener hole damage initiation model utilizes a finite element based Fourier series solution, and is validated through comparisons of analytical and experimental results.

## INTRODUCTION AND BACKGROUND

Two types of delaminations generally occur during the manufacture and assembly of composite aircraft structures; single level and multilevel delaminations. Single level delaminations are typically due to sources such as foreign objects between plies, disbonds during unbagging, thermal stresses during the cure cycle, or trapped voids. Multilevel delaminations are usually caused by an interlaminar failure at a fastener during assembly and will be addressed in this paper.

Assembly induced delaminations have been found on the AV-8B and F/A-18 aircraft. In both cases, the delaminations have occurred primarily at fasteners where gaps exist between surface skin and substructure. When the fastener is torqued up, the transverse load closes the unshimmed gap

<sup>1</sup> The analytical work described in this paper is being performed for the Naval Air Development Center and the Federal Aviation Administration under contract N62269-90-C-0281, 'Delamination Methodology for Composite Structure.'

and, if the gap is too large, creates a delamination in either the skin, the substructure, or both depending on the relative stiffnesses and strengths of the elements.

Delaminations of this type were first found on the AV-8B in the upper compression skin of the wing near the inboard pylon, Figure 1. The delaminations were caused by gaps between the skin and substructure that were not shimmed. The maximum gap condition occurred at the inboard pylon location where a composite sinewave spar intersects a composite rib with aluminum fittings (Figure 2). As shown in Figure 3, a close up view of a straight edge laid across a spar reveals a significant gap. When large gaps are not properly shimmed and fasteners are installed, delaminations will result as shown by the ultrasonic inspection of the skin at the inboard pylon (Figure 4). The delamination encompasses several fastener holes. The gap condition at this location was significantly improved by modifying the manufacturing process for the wing assembly. This included changing the procedure for using liquid shim, as well as changes in tooling and detail geometry changes in the metal and composite parts. Other issues besides unshimmed gaps at fastener holes can cause delaminations during assembly (Figure 5), but these causes were not found to be as significant as unshimmed gaps.

A methodology, which includes analysis methods to predict the post-delamination response of damaged laminates, is needed to establish criteria for acceptance, rejection, or repair of delaminated structures. McDonnell Aircraft Company (MCAIR) is currently conducting a research and development program for the Naval Air Development Center and the Federal Aviation Administration to develop such a methodology. The approach being pursued in this program is to: 1 - rely on MCAIR's existing test database to help characterize and idealize the details of typical damage; 2 - use existing analytical models for predicting local stability, strength failures, and crack growth in laminates containing idealized damage whenever possible; and 3 - develop new special purpose analyses when there are deficiencies in the existing models.

The relationship of the analysis model to the damage detected is the key to successfully predicting the response of the structure with the defect in place. An A-Scan of the delaminated area around a fastener hole in a fastener torque-up specimen is illustrated in Figure 6, and shows the damage is actually a series of elliptical delaminations stacked through the thickness of the laminate. The complex nature of fastener induced delamination is further demonstrated by the photomicrograph shown in Figure 7. Several delaminations are clearly visible and there is considerable transverse matrix cracking. Transverse matrix cracks can grow through plies until they reach an interface and then grow as a delamination, which may not extend to free surfaces.

#### EXPERIMENTAL STUDY

A test program was initiated at MCAIR to evaluate the delamination resistance and damage tolerance of several thermoset and thermoplastic material systems which show potential for use in advanced aircraft

structure. The goal was to understand the initiation of assembly induced delaminations and to identify delamination mechanisms in tough composite materials. The material systems chosen for this study were: the baseline epoxy system used on the AV-8B and F/A-18 (AS4/3501-6), a toughened epoxy (IM7/8551-7A), and a thermoplastic (AS4/PEEK). The general test plan for the AS4/PEEK and AS4/8551-7A specimen is summarized in Figure 8.

Each material system was used to manufacture test panels of equal thickness (.224 in.) and identical stacking sequence using combinations of 0,  $\pm 22.5$ ,  $\pm 45$ ,  $\pm 67.5$ , and 90 degree plies. After fabrication, the panels were machined into rectangular compression test specimens (Figure 9a) and fasteners were installed to simulate the spar/rib intersections where assembly induced delaminations frequently occur. Cross section specimens were also fabricated (Figure 9b) to evaluate the microstructure at a delamination by obtaining photomicrographs of the edge of the delamination. The delaminations were viewed at 0, 45, and 90 degree cuts through the delamination.

The test setup used to produce the delaminations is shown in Figure 10. The fastener is tightened by a torque wrench from the countersunk side of the fastener, similar to the production situation. The specimen is supported by ~3 in. diameter pipe. A load cell and a deflectometer were included in the setup so that specimen load deflection plots were obtained for each material system, as shown in Figure 11. The toughened thermoset IM7/8551-7A specimens delaminated at 32% higher load than the baseline AS4/3501-6 material, and the AS4/PEEK thermoplastic specimens delaminated at a load 67% higher than the baseline. The deflection of the thermoplastic specimens were nearly double the baseline panel when delamination occurred. This indicates that the thermoplastic material is more damage resistant and could withstand twice the unshimmed gap prior to delamination occurring.

Static strength and fatigue tests were planned on baseline (no delamination) and delaminated panels. The post delamination static compression strength of all three materials is compared to their baseline undelaminated strengths in Figure 12. The delamination size for the AS4/PEEK was larger (~2 in. dia.) than the IM7/8551-7A (~1.5 in. dia.) but the percentage reduction in compression strength from the undelaminated cases are equivalent. Compression fatigue testing with and without assembly induced delaminations is currently in progress.

#### ANALYTICAL MODELS

A thorough analytical investigation of assembly induced delaminations must address three problems; 1 - defining the conditions under which a delamination will be initiated, and the size of the initial delamination, 2 - determining the residual strength of a laminate which includes a delamination, and 3 - determining if an existing delamination will grow when the laminate is loaded. Most of the analytical work performed to date has concentrated on the second and third problems, and has considered damage due to low velocity impact events rather than assembly induced damage. In many cases, however, the models intended for low

velocity impact damage can be used to study assembly induced damage with little or no modification.

### Review of Existing Delamination Models

In general, existing delamination analyses study one or more of three basic failure modes; 1 - static failure in or near the delaminated region, 2 - buckling of one or more of the sublaminates created by the delamination, and/or 3 - delamination growth caused by static or buckling-induced loads. These three modes can work in conjunction to cause a catastrophic failure.

Most of the previous works have considered very specialized cases, assuming such things as orthotropic materials, pure compression loads, etc. In addition, much of this work considers only single level delaminations, even though impact and assembly induced damage is typically characterized by multiple level delaminations. Only a few researchers have provided fairly comprehensive discussions of more generalized analysis methods [1,2].

A large percentage of the single level delamination work involves the use of 2-D and 3-D finite element analysis to predict the onset and growth of delaminations [3-14]. These predictions are generally made through strain energy release rate calculations. The finite element analyses generally show good agreement with test data, but are relatively time-consuming to perform.

Several simplified solutions for flat laminates containing single level delaminations also exist. These analyses fall into one of three general groups; 1 - edge delamination analyses [11,15-17], 2 - through-width delamination analyses [2,18-23], and 3 - embedded delamination analyses [1,2,7,11,24-28].

Edge delaminations are generally caused by tensile loading. Since free edges of a laminate under tensile loading must be stress-free, out of plane stress concentrations develop near the edges, which can cause the plies to delaminate. Pipes and Pagano [15] presented a well known solution for these interlaminar stresses near a free edge in cross plied laminates which was later extended to more general laminates [16]. Damage growth predictions for edge delaminations have been made by Wilt, et al [11] using the finite element method; and Armanios, et al [17] developed a shear deformable plate model to predict stresses near the edge delamination tip.

Many researchers have investigated analytical models of through-width delaminations, which are used to predict sublaminates buckling and/or growth of the delamination. These models present less formidable analytical challenges since they represent what are essentially two dimensional problems, but are less representative of actual aircraft damage scenarios than edge and embedded delamination models. Whitcomb [5] used finite elements to perform parametric studies on the growth of through-width delaminations loaded into a postbuckled state and, in another paper [18], used a simplified Rayleigh-Ritz model to perform a parametric study of the stability of sublaminates which included thermal

effects. Vizzini and Lagace [19] developed a sublaminar buckling model which included the effects of an elastic foundation on the response.

Yin [20] presented a closed form solution for buckling of through-width delaminations under combined in-plane loading. Yin also presents an expression for strain energy release rate for the combined load case. Kardomateas [21] also presented a closed form solution for local and global buckling of through-width delamination under axial compression only. Martin [22] used a combination of curved beam elasticity solutions and finite element models to predict the onset and growth of delamination in unidirectional curved laminates. Sankar [23] presented a specialized beam finite element which was used to calculate strain energy release rates.

The third and most common type of delamination is an embedded delamination. Residual strength models have typically modeled embedded delaminations as elliptic inclusions in a parent (undamaged) laminate. Lekhnitskii [24] solved this problem for the stresses in and around the inclusion, and his solution has served as the basis for many subsequent models. Cairns [1,25] used Lekhnitskii's solution as part of a larger effort to predict the damage resistance and damage tolerance of laminates subject to low velocity impact. Cairns also presented an assumed-modes Rayleigh-Ritz method for predicting sublaminar buckling. Several other authors [2,7,24-28] have developed models for predicting buckling loads of elliptical delaminations making use of varying assumptions about loading, geometry, and material symmetry.

In general the delaminated sublaminates will not be symmetric or balanced and the associated coupling effects should be included in the buckling load calculations. A simple way of estimating these coupling effects is to modify the bending terms using a reduced bending stiffness (RBS) approach [29-31]. This method involves inverting the full  $6 \times 6$  matrix formed by the A, B and D matrices from laminated plate theory, to include effects of the A and B matrices in the D matrix coefficients.

When a sublaminar buckles it experiences an out-of-plane displacement which in turn induces out-of-plane loads at the perimeter of the delamination. If these loads are large enough, the delamination will grow. Various methods have been studied to predict this delamination growth. The most common of these is to calculate the strain energy release rates for an assumed delamination growth [2,5,13,14,21,32-34]. This rate is then compared with material property data to determine the load at which growth will occur. Cairns [25] suggests the use of Marguerre initial imperfection theory to calculate stresses at the boundary of the sublaminar. A strength of materials approach is then used to predict delamination growth.

Although these embedded delamination models were originally formulated for single level delaminations, they can be extended to analyze the multiple level delamination case. This is typically done by combining one of the Rayleigh-Ritz buckling analyses with Lekhnitskii's elliptical inclusion model. The delaminations are assumed to divide the laminate into several elliptical sublaminates. The strength of the delaminated region is then found by analyzing the response of each individual

sublamine. When an instability failure (sublamine buckling) is predicted in a sublamine, it can carry no additional load and in effect becomes a reduced stiffness inclusion. Any additional load must be sheared around the inclusion or redistributed to other sublaminates. If a strength failure is predicted the sublamine can carry no load, and the existing load must be redistributed to other sublaminates. Using this approach, the load is applied incrementally, and the progression of sublamine failures are tracked until all sublaminates have experienced strength failures.

### Fastener Hole Damage Initiation Model

The existing analytical models are useful for predicting the effects of delaminations on the response of uniform continuous laminates. In the case of assembly induced damage, however, delaminations typically occur around structural details such as fastener holes. Models which predict the initiation of delaminations around fastener holes have been developed [35,36], but they only consider in plane tensile loads and do not account for out of plane loads due to the fastener.

As a first step toward developing a comprehensive model for analyzing the effect of delaminations around fastener holes, MCAIR has developed a damage initiation model that considers out-of-plane loads. This model represents an annulus of orthotropic material around a countersunk fastener hole, as shown in Figure 13. The annulus is modeled in two dimensions using a special purpose anisotropic harmonic axisymmetric finite element that was developed specifically for this analysis.

The anisotropic harmonic axisymmetric element assumes that the geometry of the element is truly axisymmetric, but allows the material properties and strain fields to vary in the circumferential direction. Degrees of freedom associated with the element are translations in the radial, circumferential, and axial directions at each node ( $u$ ,  $v$ , and  $w$ , respectively). To retain generality in the circumferential variations of the displacements, they are expressed as Fourier series in  $\theta$ . The three dimensional displacement fields for the element are then given by:

$$u = u_0 + \sum_{i=1}^{\infty} (u'_i \cdot \cos(i\theta) + u''_i \cdot \sin(i\theta)) \quad (1)$$

$$v = v_0 + \sum_{i=1}^{\infty} (v'_i \cdot \sin(i\theta) - v''_i \cdot \cos(i\theta)) \quad (2)$$

$$w = w_0 + \sum_{i=1}^{\infty} (w'_i \cdot \cos(i\theta) + w''_i \cdot \sin(i\theta)) \quad (3)$$

Where  $u_0$ ,  $v_0$ ,  $w_0$ ,  $u'_i$ ,  $v'_i$ ,  $w'_i$ ,  $u''_i$ ,  $v''_i$ , and  $w''_i$  are functions of  $R$  and  $Z$  only. The primed displacement components correspond to modes that are symmetric with respect to the  $\theta = 0$  plane, and the double primed components correspond to asymmetric modes.

Since three dimensional displacement fields are assumed, all six strain components may develop. In cylindrical coordinates, these strain components are defined as:

$$\varepsilon = \begin{Bmatrix} \varepsilon_{rr} \\ \varepsilon_{\theta\theta} \\ \varepsilon_{zz} \\ \gamma_{r\theta} \\ \gamma_{\theta z} \\ \gamma_{zr} \end{Bmatrix} = \begin{Bmatrix} \frac{\partial u}{\partial r} \\ \frac{u}{r} + \frac{1}{r} \frac{\partial v}{\partial \theta} \\ \frac{\partial w}{\partial z} \\ \frac{\partial v}{\partial r} - \frac{v}{r} + \frac{1}{r} \frac{\partial u}{\partial \theta} \\ \frac{1}{r} \frac{\partial w}{\partial \theta} + \frac{\partial v}{\partial z} \\ \frac{\partial u}{\partial z} + \frac{\partial w}{\partial r} \end{Bmatrix} \quad (4)$$

Substituting (1), (2), and (3) into (4) yields a general expression for the three dimensional strain field

$$\varepsilon = \varepsilon_0 + \sum_{i=1}^{\infty} (\varepsilon'_i + \varepsilon''_i) \quad (5)$$

Where

$$\varepsilon_0 = \begin{bmatrix} \frac{\partial}{\partial r} & 0 & 0 \\ \frac{1}{r} & 0 & 0 \\ 0 & 0 & \frac{\partial}{\partial z} \\ 0 & \frac{1}{r} - \frac{\partial}{\partial r} & 0 \\ 0 & -\frac{\partial}{\partial z} & 0 \\ \frac{\partial}{\partial z} & 0 & \frac{\partial}{\partial r} \end{bmatrix} \begin{Bmatrix} u_0 \\ v_0 \\ w_0 \end{Bmatrix} \quad (6)$$

$$\varepsilon'_i = \begin{bmatrix} \cos(i\theta) \frac{\partial}{\partial r} & 0 & 0 \\ \cos(i\theta) \frac{1}{r} & \cos(i\theta) \frac{i}{r} & 0 \\ 0 & 0 & \cos(i\theta) \frac{\partial}{\partial z} \\ -\sin(i\theta) \frac{i}{r} & -\sin(i\theta) \left( \frac{1}{r} - \frac{\partial}{\partial r} \right) & 0 \\ 0 & \sin(i\theta) \frac{\partial}{\partial z} & -\sin(i\theta) \frac{i}{r} \\ \cos(i\theta) \frac{\partial}{\partial z} & 0 & \cos(i\theta) \frac{\partial}{\partial r} \end{bmatrix} \begin{Bmatrix} u'_i \\ v'_i \\ w'_i \end{Bmatrix} \quad (7)$$



and

$$\varepsilon''_i = \begin{bmatrix} \sin(i\theta) \frac{\partial}{\partial r} & 0 & 0 \\ \sin(i\theta) \cdot \frac{1}{r} & \sin(i\theta) \cdot \frac{i}{r} & 0 \\ 0 & 0 & \sin(i\theta) \cdot \frac{\partial}{\partial z} \\ \cos(i\theta) \cdot \frac{i}{r} & \cos(i\theta) \cdot \left( \frac{1}{r} - \frac{\partial}{\partial r} \right) & 0 \\ 0 & -\cos(i\theta) \cdot \frac{\partial}{\partial z} & \cos(i\theta) \cdot \frac{i}{r} \\ \sin(i\theta) \cdot \frac{\partial}{\partial z} & 0 & \sin(i\theta) \cdot \frac{\partial}{\partial r} \end{bmatrix} \begin{Bmatrix} u''_i \\ v''_i \\ w''_i \end{Bmatrix} \quad (8)$$

The geometry of the element is defined by twelve nodes in the R-Z plane, which represent an annulus of material, as shown in Figure 14. Displacements throughout the domain of the element are approximated using interpolation functions of the form

$$f(r,z) = \sum_{n=1}^{12} h_n(r,z) \cdot f_n \quad (9)$$

Where the  $h_n$  are coefficients in the cubic interpolation function for an ordinary two dimensional 'serendipity' type element, and  $f_n$  is the displacement at node  $n$ .

Approximating  $u_0$ ,  $v_0$ ,  $w_0$ ,  $u'_i$ ,  $v'_i$ ,  $w'_i$ ,  $u''_i$ ,  $v''_i$ , and  $w''_i$  using (9), and substituting into (6), (7), and (8) provides an expression for the strain throughout the element in terms of the nodal displacements.

$$\varepsilon = B_0 \begin{Bmatrix} u_{01} \\ v_{01} \\ w_{01} \\ \vdots \\ u_{012} \\ v_{012} \\ w_{012} \end{Bmatrix} + \sum_{i=1}^{\infty} B'_i \begin{Bmatrix} u'_{i1} \\ v'_{i1} \\ w'_{i1} \\ \vdots \\ u'_{i12} \\ v'_{i12} \\ w'_{i12} \end{Bmatrix} + B''_i \begin{Bmatrix} u''_{i1} \\ v''_{i1} \\ w''_{i1} \\ \vdots \\ u''_{i12} \\ v''_{i12} \\ w''_{i12} \end{Bmatrix} = B \cdot u \quad (10)$$

The constitutive law for the laminates considered in this program will be constant when expressed in rectangular coordinates, and is defined by

$$\sigma^* = D^* \cdot \varepsilon^* \quad (11)$$

where

$$\varepsilon^* = \begin{Bmatrix} \varepsilon_{xx} \\ \varepsilon_{yy} \\ \varepsilon_{zz} \\ \gamma_{xy} \\ \gamma_{yz} \\ \gamma_{zx} \end{Bmatrix}, \quad \sigma^* = \begin{Bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \sigma_{zz} \\ \tau_{xy} \\ \tau_{yz} \\ \tau_{zx} \end{Bmatrix} \quad (12,13)$$

and  $\mathbf{D}^*$  is a 6 X 6 matrix that can be, in general, fully populated. The constitutive law in cylindrical coordinates is calculated from (11), using the strain transformation defined by

$$\varepsilon = \mathbf{T} \cdot \varepsilon^* \quad (14)$$

where

$$\mathbf{T} = \begin{bmatrix} \cos^2\theta' & \sin^2\theta' & 0 & \sin\theta'\cos\theta' & 0 & 0 \\ \sin^2\theta' & \cos^2\theta' & 0 & -\sin\theta'\cos\theta' & 0 & 0 \\ 0 & 0 & 1 & 0 & 0 & 0 \\ -2\sin\theta'\cos\theta' & 2\sin\theta'\cos\theta' & 0 & \cos^2\theta' - \sin^2\theta' & 0 & 0 \\ 0 & 0 & 0 & 0 & \cos\theta' & -\sin\theta' \\ 0 & 0 & 0 & 0 & \sin\theta' & \cos\theta' \end{bmatrix} \quad (15)$$

and

$$\theta' = \theta - \beta \quad (16)$$

Substituting (14) and a similar transformation for stress into (11) yields

$$\sigma = \mathbf{D} \cdot \varepsilon \quad (17)$$

where

$$\sigma = \begin{Bmatrix} \sigma_{rr} \\ \sigma_{\theta\theta} \\ \sigma_{zz} \\ \tau_{r\theta} \\ \tau_{\theta z} \\ \tau_{zr} \end{Bmatrix} \quad (18)$$

and  $\mathbf{D}$  is a  $\theta$  dependent expression given by

$$\mathbf{D} = \mathbf{T}^T \cdot \mathbf{D}^* \cdot \mathbf{T} \quad (19)$$

Using standard finite element procedures [37], the effective stiffness of the element is calculated by minimizing the potential energy, which is defined in terms of the nodal unknowns using (10) and (17).

$$\mathbf{K} = \int_{\text{vol}} \mathbf{B}^T \cdot \mathbf{D} \cdot \mathbf{B} \, dv \quad (20)$$

Unlike conventional axisymmetric and isotropic or cylindrically orthotropic harmonic axisymmetric elements, integrating (20) in the circumferential direction is not trivial, and must be performed numerically. In addition, the anisotropic nature of  $\mathbf{D}$  leads to coupling between the modes of the Fourier series that does not exist for isotropic or cylindrically orthotropic elements. Each element in a model that considers  $j$  modes will then have  $(36 + 72 \cdot j)$  degrees of freedom, consisting of  $u_0$ ,  $v_0$ , and  $w_0$  at each node, and  $u'_j$ ,  $v'_j$ ,  $w'_j$ ,  $u''_j$ ,  $v''_j$ , and  $w''_j$  for each mode at each node.

Since the element uses cubic interpolation functions in the R-Z plane, the geometry shown in Figure 13 can be modeled with a relatively coarse mesh. The number of terms in the Fourier series required to characterize circumferential variations depends on the degree of anisotropy of  $\mathbf{D}^*$  and on the applied load. Two different load cases can be considered, one represents a properly seated fastener and the other represents an improperly seated fastener (Figure 15). For the properly seated case the normal pressure on the countersink is assumed to be constant in  $\theta$ , and for the improperly seated case it is defined by

$$P(\theta) = P \cdot (1 + \cos(\theta)) \quad (21)$$

The number of terms in the Fourier series for various material types and loads cases is listed in Table 1. In practice, the solution predicted by this model converges very rapidly, and only a few terms in the Fourier series are required. A convergence study was performed for a .1040 inch thick AS4/3501-6 laminate with a  $[(\pm 67.5)_2/90/45/(-45)_2/(0)_2]_S$  stacking sequence. This study showed that the displacement solution converged to within .001% using only three terms in the series.

Results predicted by the model have been compared to some of the experimental results described above. Comparisons of the analytically and experimentally determined effective stiffnesses for AS4/3501-6, IM7/8551-7A, and AS4/PEEK laminates are shown in Figure 16. The geometrical parameters and stacking sequence used for these analyses are listed in Table 2. For these comparisons, the effective stiffness of each laminate was taken as the axial load in the fastener divided by the maximum lateral deflection of the laminate. As shown in Figure 16, there is very good agreement between the experimental and analytical results.

The initiation of damage in the AS4/3501-6 laminate was also predicted analytically, based on the average stresses calculated over a characteristic length away from the fastener hole. For this analysis, it

was assumed that ply interfaces were isotropic matrix rich regions, and that a delamination would occur when the von Mises stress in an interface exceeded the shear strength of the matrix. Since the stress and strain fields are three dimensional, each interface was searched in the circumferential direction to identify the critical location for the initiation of damage, and the radial length of the element closest to the fastener hole was used as the characteristic length. The predicted damage initiation load as a function of the characteristic length is shown in Figure 17, along with experimental data for the same laminate. The properly and improperly seated cases bracket the experimental data very well, and the properly seated case is tending toward the experimental result as the characteristic length approaches zero. The failure predicted by the analysis was at a circumferential location approximately  $45^\circ$  away from the  $0^\circ$  direction of the laminate and just below the bottom of the countersink, which corresponds well with the observed failure.

## DISCUSSION AND CONCLUSIONS

The results of the experimental study indicate that it may be possible to improve the resistance of aircraft to assembly induced delaminations by using a tough epoxy or thermoplastic material system. The fastener loads required to cause delaminations in these systems were 32% and 67% higher than the initial delamination load for the conventional epoxy system. The fact that the toughened epoxy and thermoplastic specimens experienced larger lateral deflections before delaminating implies that structure fabricated from these materials can tolerate larger unshimmed gaps than similar structures fabricated from conventional epoxy. Strength reductions due to the presence of delaminations in the tougher systems were comparable to the reduction observed for conventional epoxy specimens, although the thermoplastic does start from a baseline strength that is below that for either thermoset system.

The survey of existing analytical methods shows that while a significant amount of research has been conducted to develop models for studying delaminations in general, very little work has been done on the specific problem of analyzing delaminations around fastener holes. The existing sublaminar buckling and elliptical inclusion models for analyzing the response of a laminate with an embedded delamination subject to in-plane loads may be used to study the effects of delaminations around fastener holes. However, they must be modified to account for the additional constraint the fastener places on the buckled mode shape, and for the stress concentration due to the fastener hole. The existing fracture mechanics approaches to predict the growth of these delaminations will require similar modifications.

Finite element based solutions appear to be the most appropriate methods for analyzing fastener induced delaminations subject to out-of-plane loads, such as those due to the fastener itself. The anisotropic harmonic axisymmetric element developed at MCAIR provides a means for efficiently modeling this problem. This approach has been demonstrated to accurately predict both the location and load required to produce the

initial delamination. This same model can be used to explicitly model delaminations by treating the elliptical delamination as an effective circular delamination. It may also be possible to use this model to numerically determine strain energy release rates as part of a methodology to predict the growth of an existing delamination.

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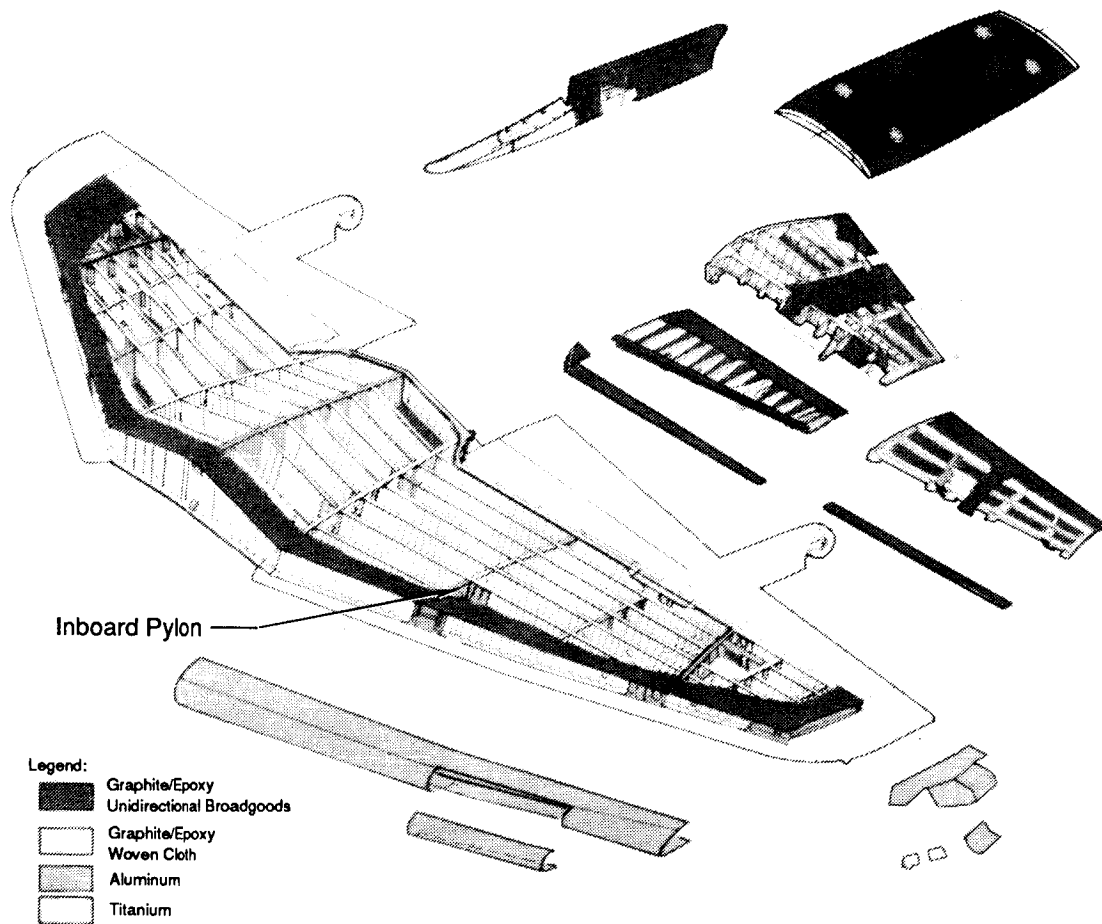
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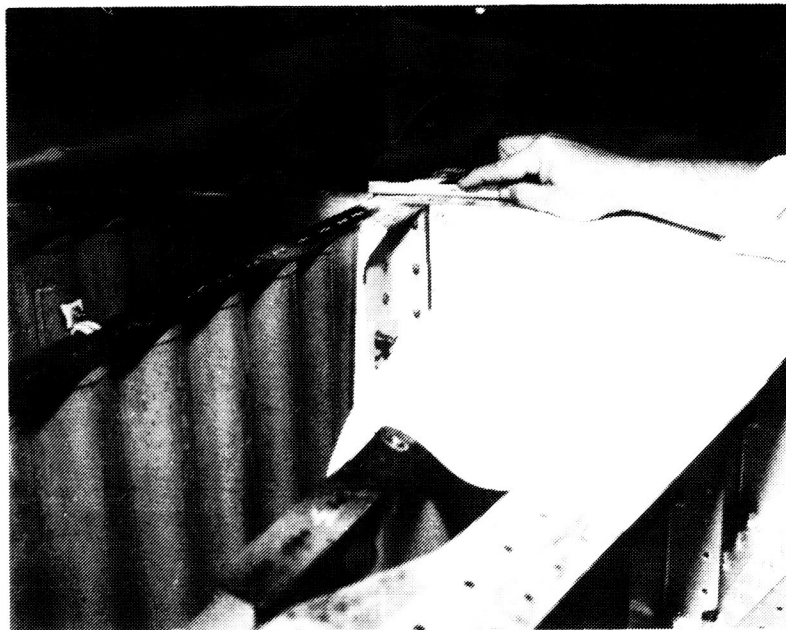
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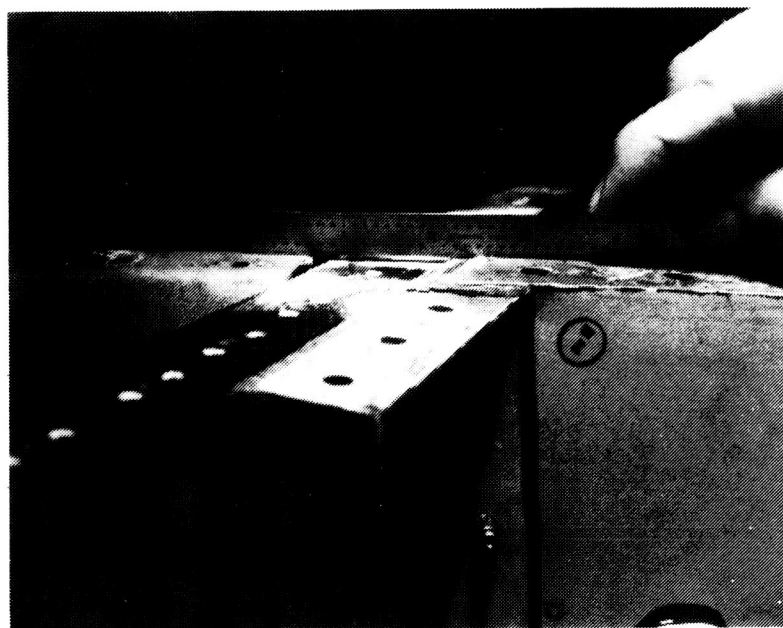
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**Figure 1. AV-8B Composite Wing**



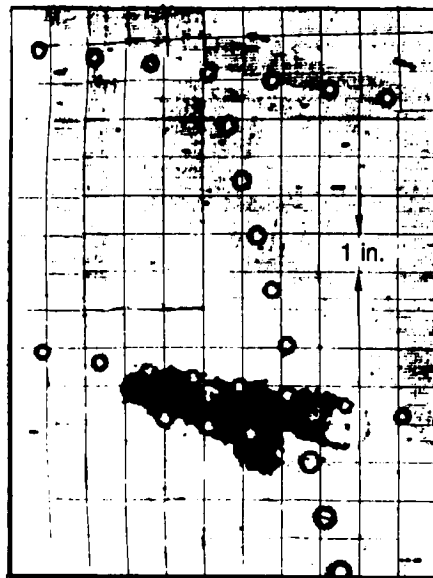
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**Figure 2. Substructure at Inboard Pylon**



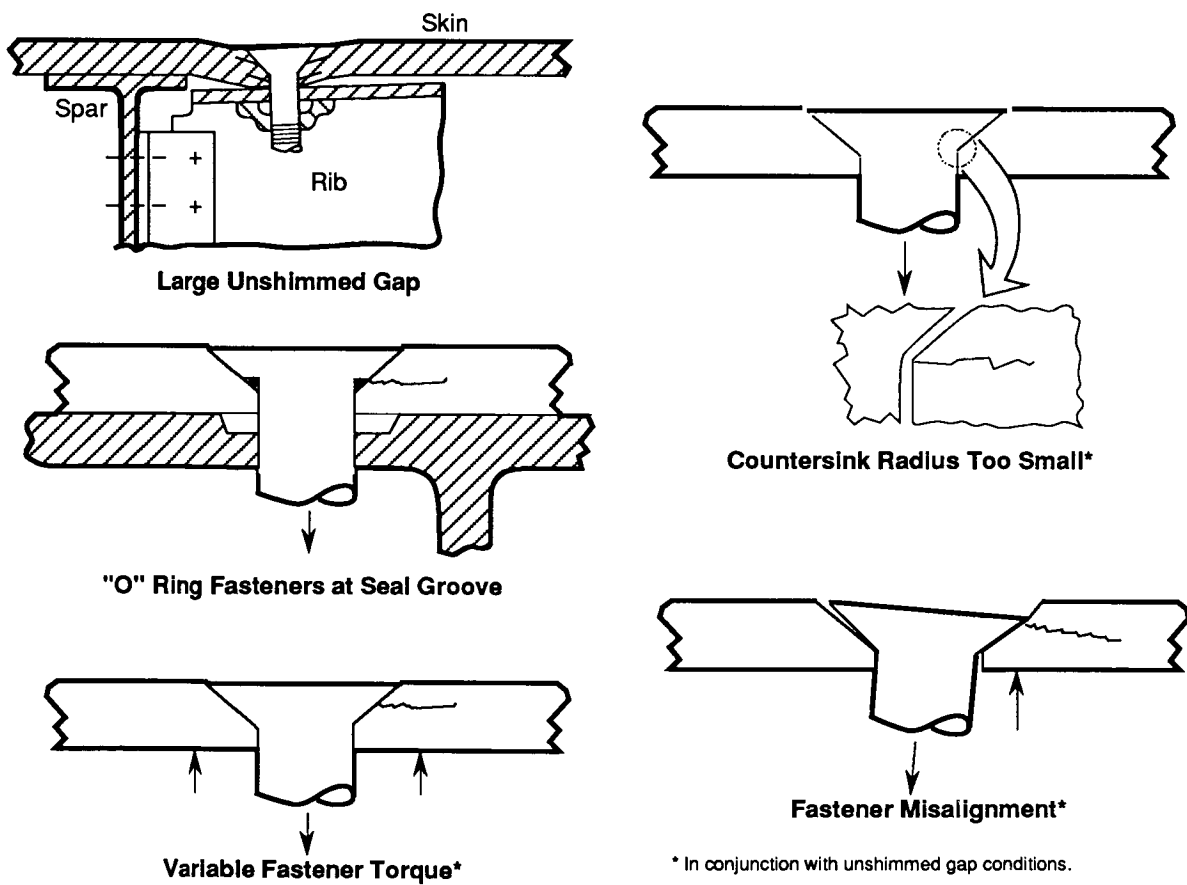
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**Figure 3. Close Up of Substructure at Inboard Pylon**



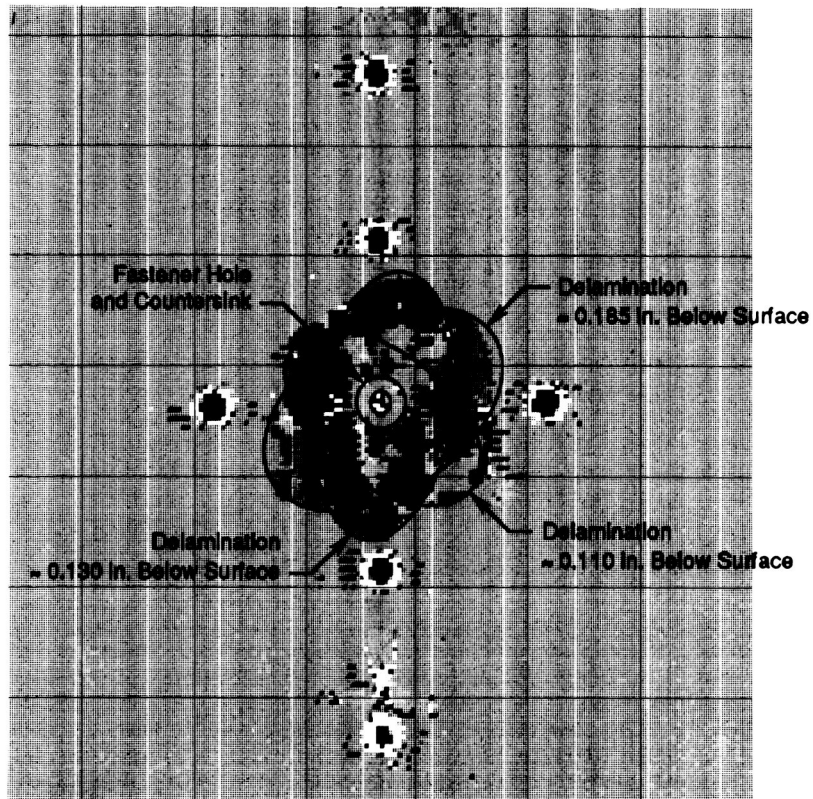
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**Figure 4. Ultrasonic Portrait of Skin Delamination**



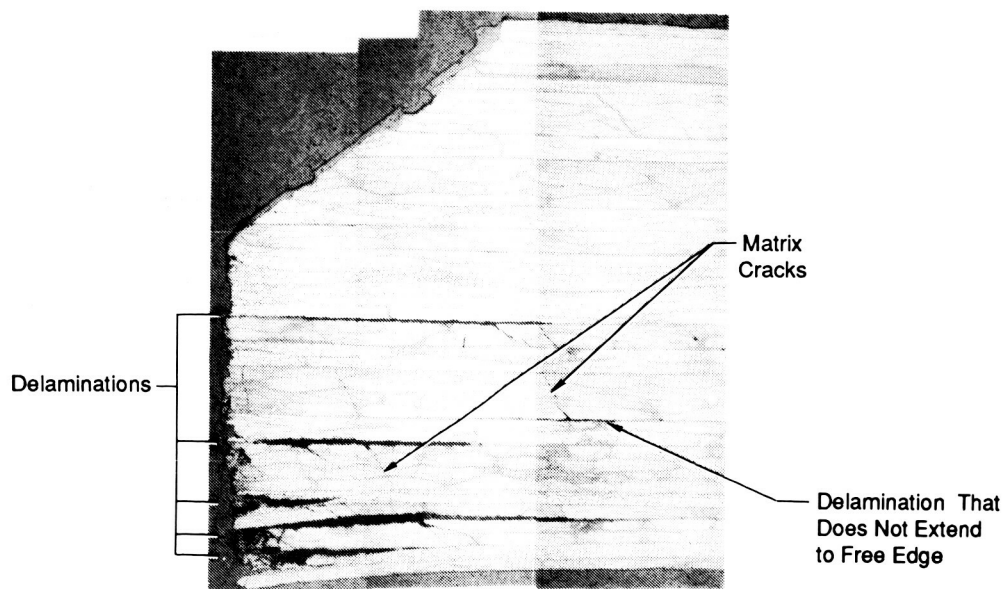
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**Figure 5. Combination of Factors Caused Delaminations at Fasteners**



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**Figure 6. A-Scan of Typical Fastener Induced Delamination**



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**Figure 7. Photomicrograph of Typical Fastener Induced Delamination**

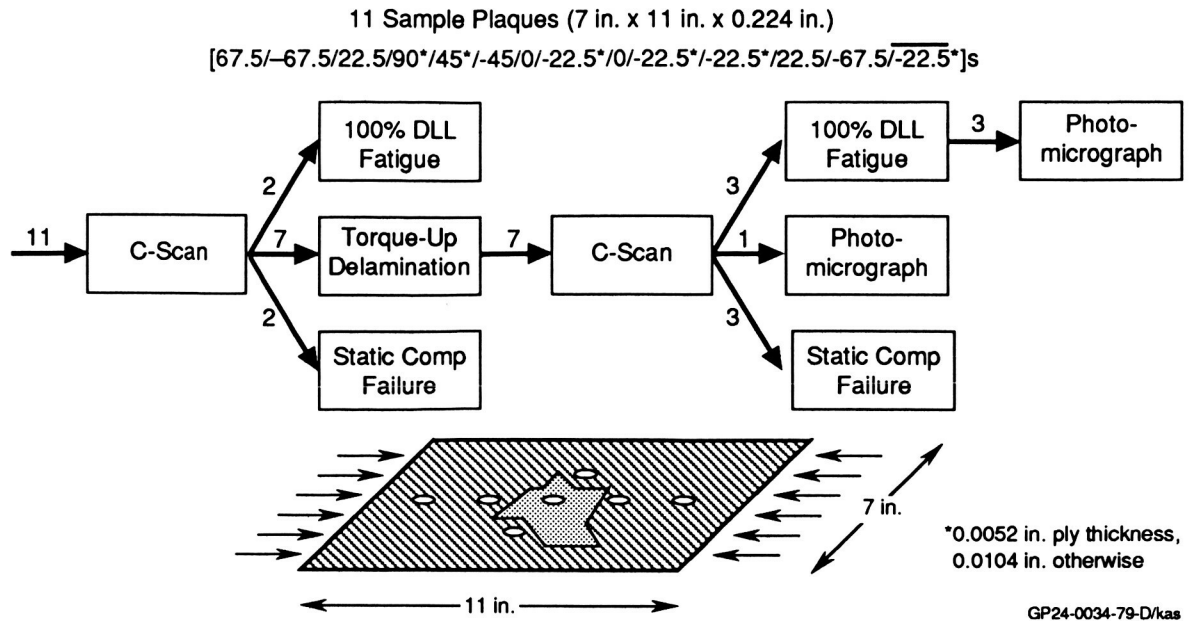


Figure 8. AS4/PEEK and AS4/8551 Test Plan

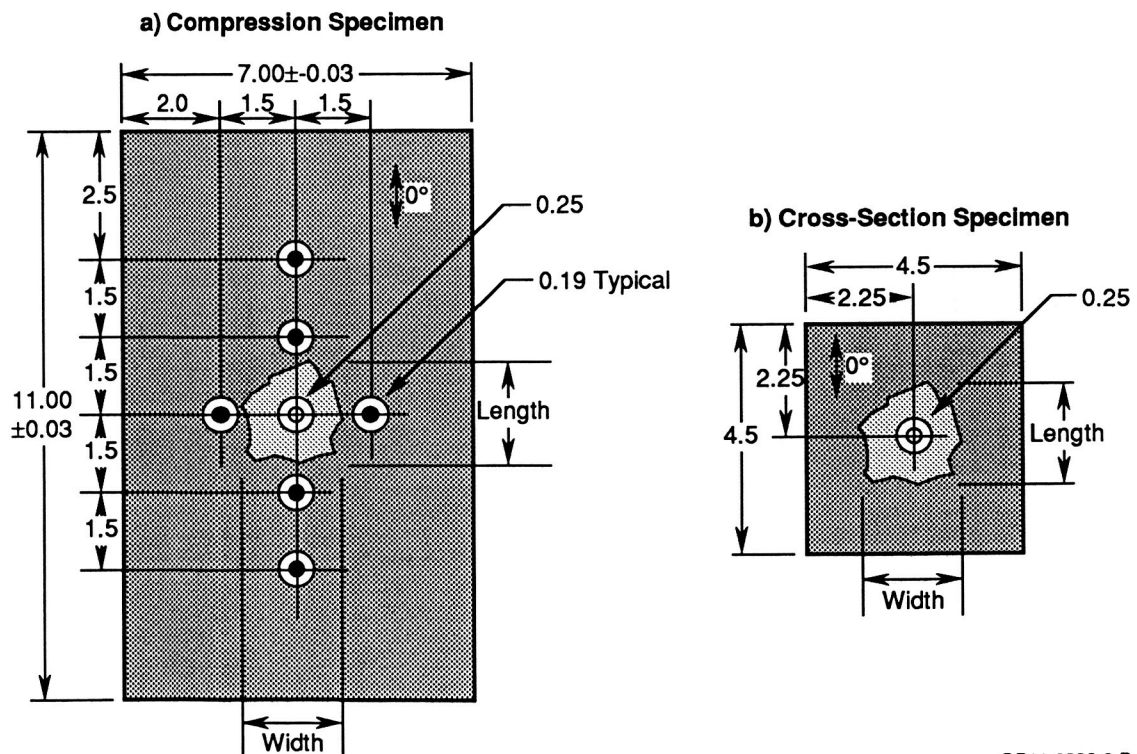
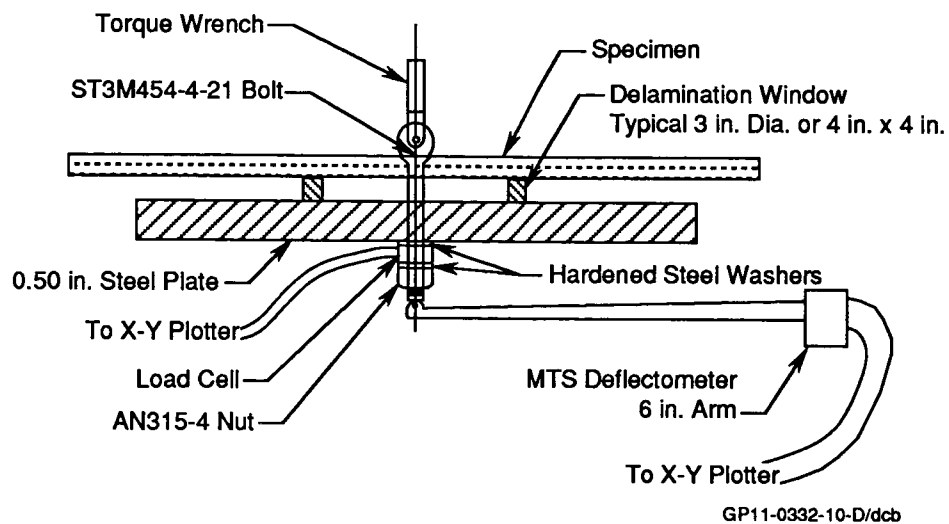
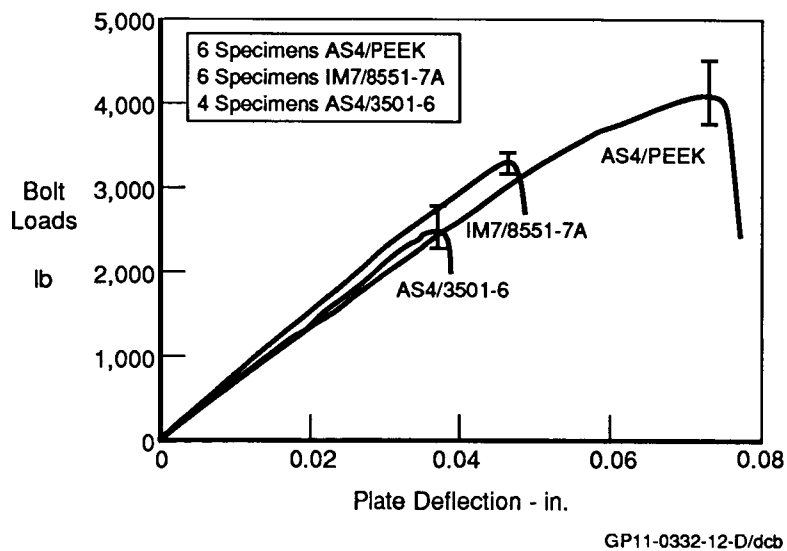


Figure 9. Test Specimens



**Figure 10. Delamination Setup**



**Figure 11. Typical Transverse Load vs Plate Deflection Plots**

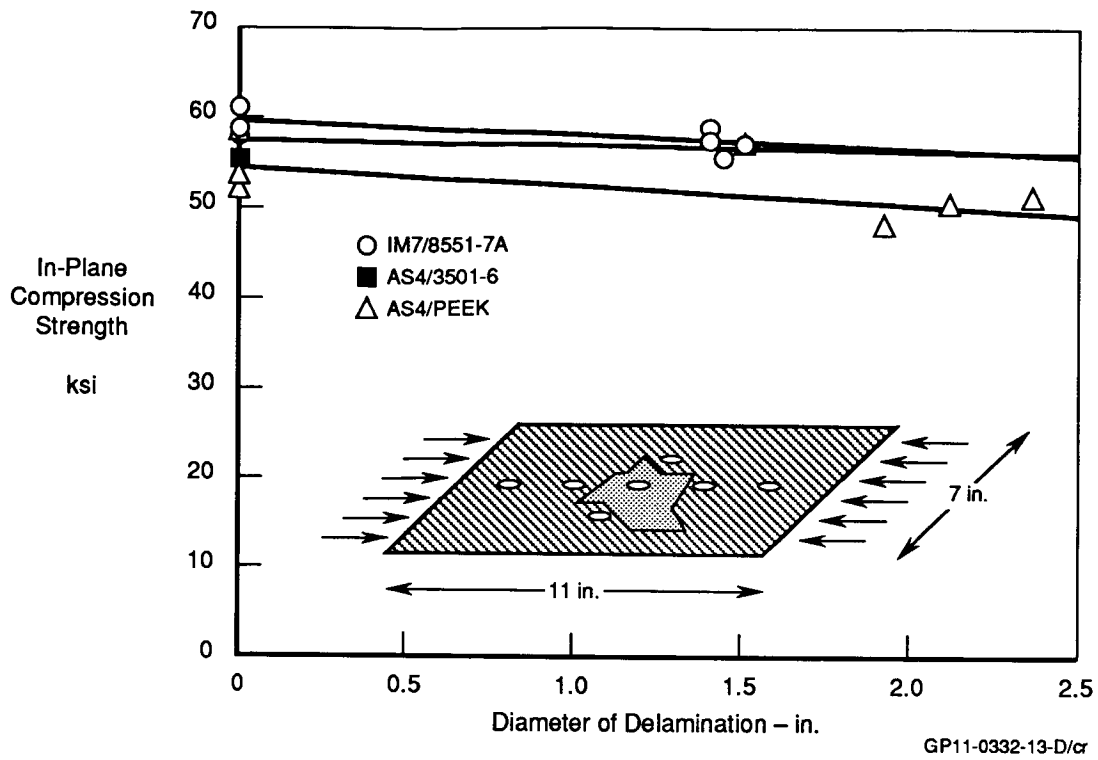


Figure 12. Comparison of the Post Delamination Compression Strength

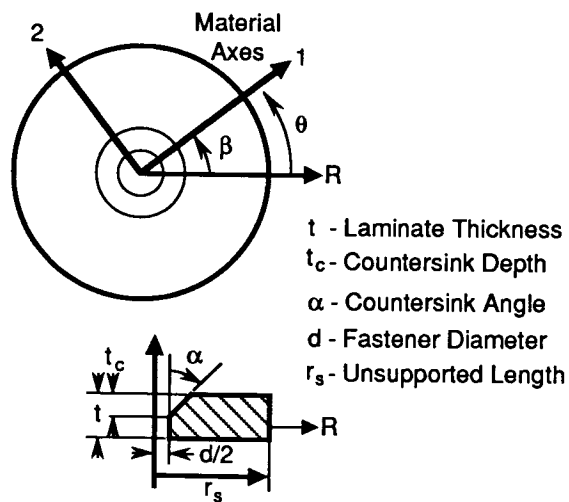
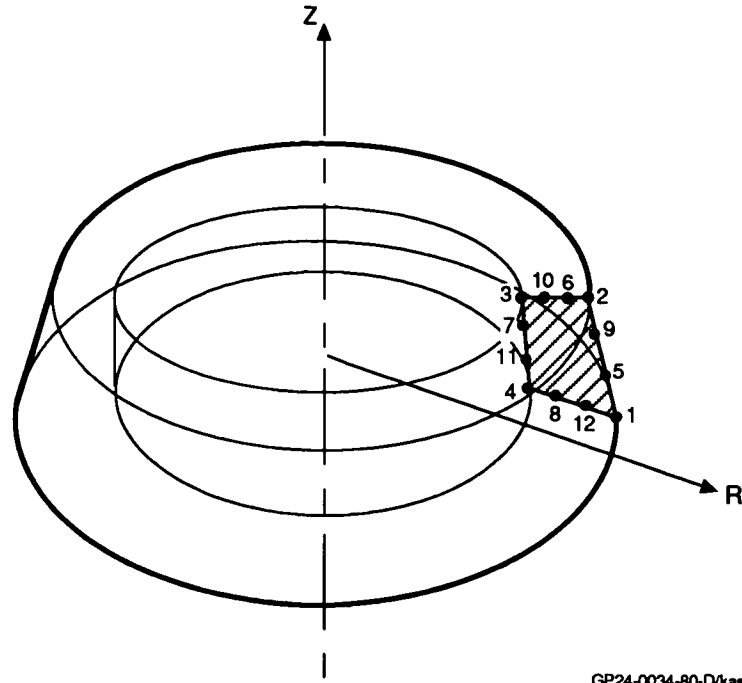
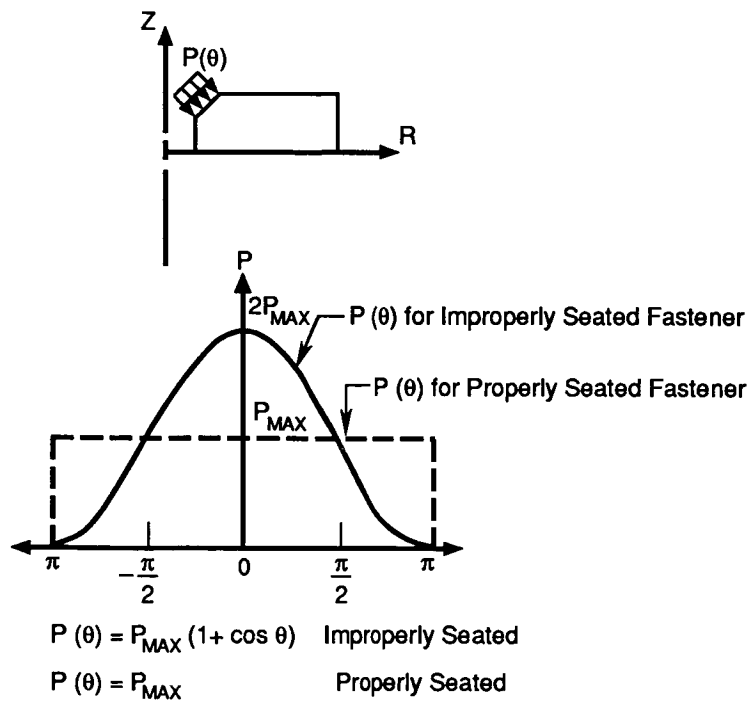


Figure 13. Fastener Hole Analysis Model Nomenclature



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**Figure 14. Node Locations for the Anisotropic Harmonic Axisymmetric Element**



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**Figure 15. Load Cases for the Fastener Hole Analysis Model**



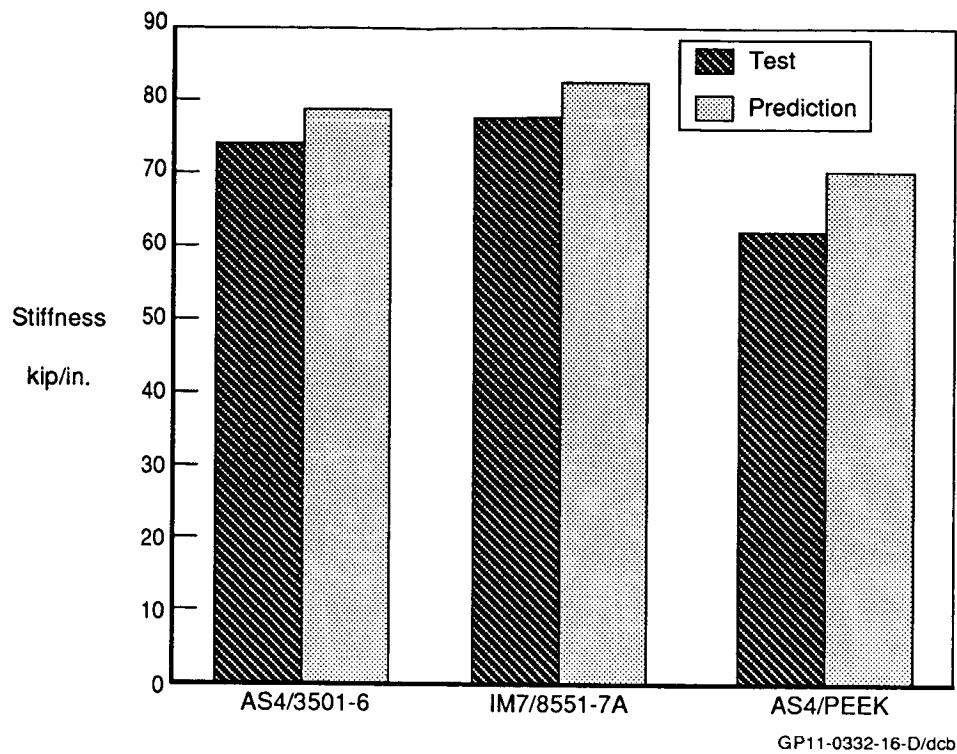


Figure 16. Effective Stiffness Comparisons

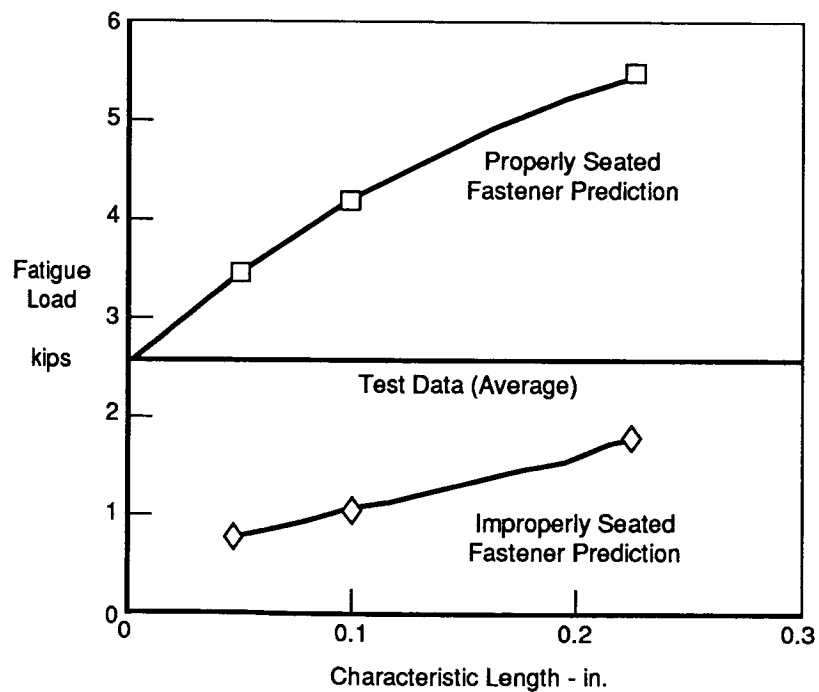


Figure 17 Initial Delamination Load Comparison  
for an AS4/3501-6 Laminate

**TABLE 1. MODES REQUIRED FOR VARIOUS ANALYSIS CASES**

| Case | D*          | Fastener Seating | $\beta$  | Required Modes  |
|------|-------------|------------------|----------|-----------------|
| I    | Isotropic   | Proper           | —        | 0               |
| II   | Isotropic   | Improper         | —        | 0 and 1st Sym.  |
| III  | Orthotropic | Proper           | —        | 0 and Even Sym. |
| IV   | Orthotropic | Improper         | 0        | 0 and All Sym.  |
| V    | Orthotropic | Improper         | $\neq 0$ | All             |

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**TABLE 2. PARAMETERS FOR THE ANALYSIS VERIFICATION MODEL**

| Parameter            | Value  |
|----------------------|--------|
| d (in.)              | 0.25   |
| t <sub>c</sub> (in)  | 0.1080 |
| $\alpha$ (°)         | 50     |
| t (in.)              | 0.2236 |
| r <sub>s</sub> (in.) | 1.5    |

Stacking Sequence -  
 $[\pm 67.5/22.5/90^{\circ}/45^{\circ}/-45/0/(-22.5^{\circ})_2/22.5/-67.5/22.5^{\circ}]_s$   
 Ply Thickness = 0.0104 in.  
 \* Ply Thickness = 0.0052 in.

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# STRESS ANALYSIS AND FAILURE OF AN INTERNALLY PRESSURIZED

## COMPOSITE-JACKETED STEEL CYLINDER

514-39

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51326

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### SUMMARY

This paper presents a nonlinear stress analysis of a thick-walled compound tube subjected to internal pressure. The compound tube is constructed of a steel liner and a graphite-bismaleimide outer shell. Analytical expressions for the stresses, strains, and displacements are derived for all loading ranges up to failure. Numerical results for the stresses and the maximum value that the compound tube can contain without failure are presented.

### INTRODUCTION

Weight reduction is a requirement for a majority of weapon systems being developed by the Army. The Army would like to design a longer cannon and maintain the inertia characteristics of the shorter cannon. This accomplishment would allow current cannon mounts to be used. The longer cannon is expected to achieve higher muzzle velocity and greater accuracy than the standard cannon. The design under consideration is to replace a portion of the steel wall thickness with a lighter material. The inner portion, the steel liner, maintains the tube projectile interface and shields the composite from the extremely hot gases. The outer portion, the composite jacket, is made of a fiber-reinforced organic composite (graphite fiber and a bismaleimide matrix). A linear stress analysis for this problem under internal pressure in the elastic range was reported in a recent paper by M.D. Witherell and M.A. Scavullo (ref. 1).

This paper presents a nonlinear stress and failure analysis of the compound tube problem. The loading ranges include elastic, elastic-plastic, and fully-plastic up to failure. Analytical expressions for the stresses, strains, and displacements are derived for all cases. Numerical results for the radial and hoop stresses in the nonlinear loading ranges are presented. The maximum value of internal pressure that the compound tube can contain without failure is predicted.

### PROBLEM AND ELASTIC ANALYSIS

Figure 1 shows a schematic of the compound tube problem. The compound tube consists of an inner steel "liner" and an outer composite "jacket." The steel liner of inside radius  $a$  and outer radius

b is wrapped in the circumferential direction with a graphite-bismaleimide organic composite of outside radius c. The elastic material constants for the composite and the steel are given in Table I.

TABLE I. ELASTIC CONSTANTS OF COMPOSITE JACKET AND STEEL LINER

| Elastic Constants for IM6/Bismaleimide, 55% F.V.R. |                           |                           |
|----------------------------------------------------|---------------------------|---------------------------|
| $E_r = 1.126 \text{ Mpsi}$                         | $\nu_{r\theta} = 0.01524$ | $\nu_{\theta r} = 0.3155$ |
| $E_\theta = 23.31 \text{ Mpsi}$                    | $\nu_{\theta z} = 0.3155$ | $\nu_{z\theta} = 0.01524$ |
| $E_z = 1.126 \text{ Mpsi}$                         | $\nu_{zr} = 0.3991$       | $\nu_{rz} = 0.3911$       |
| Elastic Constants for Steel                        |                           |                           |
| $E = 30.0 \text{ Mpsi}$                            | $\nu = 0.3$               |                           |

When the composite tube is subjected to internal pressure p in the elastic range, the general solutions in the plane-strain condition for the isotropic liner ( $a \leq r \leq b$ ) are

$$\sigma_r \quad (1)$$

$$\sigma_\theta = \left\{ \mp(p-q)\left(\frac{b}{r}\right)^2 + p - q b^2/a^2 \right\} / \left( \frac{b^2}{a^2} - 1 \right) \quad (2)$$

$$u/r = E^{-1}(1+\nu)[(p-q)(b/r)^2 + (1-2\nu)(p-q b^2/a^2)] / \left( \frac{b^2}{a^2} - 1 \right) \quad (3)$$

and for the orthotropic jacket ( $b \leq r \leq c$ ),

$$\sigma_r = q \left[ - \left( \frac{c}{b} \right)^{k-1} \left( \frac{c}{r} \right)^{k+1} + \left( \frac{r}{b} \right)^{k-1} \right] / \left[ (c/b)^{2k} - 1 \right] \quad (4)$$

$$\sigma_\theta = kq \left[ (c/b)^{k-1} \left( \frac{c}{r} \right)^{k+1} + \left( \frac{r}{b} \right)^{k-1} \right] / \left[ (c/b)^{2k} - 1 \right] \quad (5)$$

$$u/r = \epsilon_\theta = \alpha_{12}\sigma_r + \alpha_{22}\sigma_\theta \quad (6)$$

where q is the pressure at the interface,  $k = (\alpha_{11}/\alpha_{22})^{1/2}$ ,

$$\alpha_{11} = (1-\nu_{rz}\nu_{zr})/E_r$$

$$\alpha_{12} = -(\nu_{\theta r} + \nu_{\theta z}\nu_{zr})/E_\theta$$

$$\alpha_{22} = (1-\nu_{\theta z}\nu_{z\theta})/E_\theta \quad (7)$$

By requiring the displacement to be continuous at the interface, the interface pressure q can be expressed as a linear function of internal pressure p,

$$\frac{2p}{q} = \left( \frac{b^2}{a^2} - 1 \right) \left[ Ak \frac{(c/b)^{2k} + 1}{(c/b)^{2k} - 1} + B \right] + \frac{b^2}{a^2} + 1 \quad (8)$$

where

$$A = E\alpha_{22}/(1-\nu^2), \quad B = -E\alpha_{12}/(1-\nu^2) - \nu/(1-\nu) \quad (9)$$

Now all the stresses, strains, and displacements in the tube ( $a \leq r \leq c$ ) can be determined as functions of  $p$ . In particular, the expressions for the displacements at the bore ( $u_a$ ), interface ( $u_b$ ), and outside surface ( $u_c$ ) are

$$\left(\frac{b^2}{a^2} - 1\right) \frac{E}{p} \frac{u_a}{a} = (1+\nu) \frac{b^2}{a^2} + (1-\nu-2\nu^2) - \frac{4(1-\nu^2)(b^2/a^2)}{\left(\frac{b^2}{a^2} - 1\right) \left[ AK \frac{(c/b)^{2k+1}}{(c/b)^{2k-1}} + B \right] + \frac{b^2}{a^2} + 1} \quad (10)$$

$$\frac{u_b}{b} = q \left[ k\alpha_{22} \frac{(c/b)^{2k} + 1}{(c/b)^{2k} - 1} - \alpha_{12} \right] \quad (11)$$

$$\frac{u_c}{c} = \frac{2qk\alpha_{22}(c/b)^{k-1}}{(c/b)^{2k} - 1} \quad (12)$$

#### ELASTIC-PLASTIC ANALYSIS

When the internal pressure  $p$  is large enough, part of the steel liner will become plastic. Using Tresca's yield criterion, the associated flow rule, and assuming linear strain-hardening, the elastic-plastic solution based on Bland can be used (ref. 2 or ref. 3). Let  $\rho$  be the elastic-plastic interface.

The solution can be written in the elastic portion ( $\rho \leq r \leq b$ ) as

$$\frac{E}{\sigma_0} \frac{u}{r} = \frac{1+\nu}{2} \frac{\rho^2}{r^2} + (1-\nu-2\nu^2) \left[ \frac{1}{2} \frac{\rho^2}{b^2} - \frac{q}{\sigma_0} \right] \quad (13)$$

$$\frac{\sigma_r}{\sigma_0} = \frac{1}{2} \left( \mp \frac{\rho^2}{r^2} + \frac{\rho^2}{b^2} \right) - \frac{q}{\sigma_0} \quad (14)$$

$$\frac{\sigma_\theta}{\sigma_0} = \frac{1}{2} \left( \mp \frac{\rho^2}{r^2} + \frac{\rho^2}{b^2} \right) - \frac{q}{\sigma_0} \quad (15)$$

$$\frac{\sigma_z}{\sigma_0} = \nu \frac{\rho^2}{b^2} - 2\nu \frac{q}{\sigma_0} \quad (16)$$

and in the plastic portion ( $a \leq r \leq \rho$ )

$$\frac{E}{\sigma_0} \frac{u}{r} = (1-\nu-2\nu^2) \frac{\sigma_r}{\sigma_0} + (1-\nu^2) \frac{\rho^2}{r^2} \quad (17)$$

$$\frac{\sigma_r}{\sigma_0} = \mp \frac{1}{2} (1-\eta\beta+\eta\beta \frac{\rho^2}{r^2}) + \frac{1}{2} \frac{\rho^2}{b^2} - (1-\eta\beta) \ln \frac{\rho}{r} - \frac{q}{\sigma_0} \quad (18)$$

$$\frac{\sigma_\theta}{\sigma_0} = \mp \frac{1}{2} (1-\eta\beta+\eta\beta \frac{\rho^2}{r^2}) + \frac{1}{2} \frac{\rho^2}{b^2} - (1-\eta\beta) \ln \frac{\rho}{r} - \frac{q}{\sigma_0} \quad (19)$$

$$\sigma_z/\sigma_0 = \nu \rho^2/b^2 - 2\nu(1-\eta\beta)\ln \frac{\rho}{r} - 2\nu q/\sigma_0 \quad (20)$$

$$\bar{\epsilon}^\rho = \beta(\rho^2/r^2 - 1) \quad , \quad \eta\beta = \frac{m}{m + \frac{3}{4} \frac{(1-m)}{(1-\nu)^2}} \quad (21)$$

$$\eta = \frac{2}{\sqrt{3}} \frac{E}{\sigma_0} \frac{m}{1-m} \quad , \quad m = \frac{E_t}{E} \quad , \quad \sigma = \sigma_0(1+\eta\epsilon^\rho) \quad (22)$$

where  $\sigma_0$  is the initial tensile yield stress, and  $E_t$  is the tangent modulus in the plastic range of the stress-strain curve.

Using Eqs. (11) and (13) and the requirement of displacement continuity at the interface, i.e.,  $u_{b-}$  (liner) =  $u_{b+}$  (jacket), we obtain the expression for the interface pressure  $q$  as

$$\frac{q}{\sigma_0} = \frac{(1-\nu^2)\rho^2/b^2}{(1+\nu)(1-2\nu) + E[\alpha_{22}k \frac{(c/b)^{2k} + 1}{(c/b)^{2k} - 1} - \alpha_{12}]} \quad (23)$$

Given any value of  $\rho$  in  $a \leq \rho \leq b$ , we can now determine  $q$ ,  $u$ , and all the stresses and strains in the tube. In particular, the expressions for internal pressure and for displacements at the bore and the interface are

$$\frac{p_-}{\sigma_0} = \frac{q_-}{\sigma_0} + \frac{1}{2} \left(1 - \frac{\rho^2}{b^2}\right) + (1-\eta\beta)\ln \frac{\rho}{a} + \frac{1}{2} \eta\beta \left(\frac{\rho^2}{a^2} - 1\right) \quad (24)$$

$$\frac{E_-}{\sigma_0} \frac{u_a}{a} = - (1-\nu-2\nu^2) \frac{p_-}{\sigma_0} + (1-\nu^2)\rho^2/a^2 \quad (25)$$

$$\frac{E_-}{\sigma_0} \frac{u_b}{b} = (1-\nu^2) \frac{\rho^2}{b^2} - (1-\nu-2\nu^2) \frac{q_-}{\sigma_0} \quad (26)$$

By letting  $\rho = a$  and  $b$ , we can determine the lower limits  $p^*$ ,  $q^*$ ,  $u_a^*$ ,  $u_b^*$ ,  $u_c^*$ , and the upper limits  $p^{**}$ ,  $q^{**}$ ,  $u_a^{**}$ ,  $u_b^{**}$ , and  $u_c^{**}$ , respectively.

#### FULLY-PLASTIC ANALYSIS

When the internal pressure  $p$  is further increased, i.e.,  $p > p^{**}$ , the steel liner will become fully-plastic. The composite jacket remains elastic as long as the failure pressure is not reached. Using Tresca's yield criterion, the associated flow rule, and assuming linear strain-hardening, a fully-plastic solution can be obtained (ref. 4). The result is presented here for completeness. The explicit expressions for the displacement, strains, and stresses in the plane-strain case, subject to  $\sigma_\theta \geq \sigma_z \geq \sigma_r$ , are

$$ru = E^{-1}(1-2\nu)(1+\nu)r^2\sigma_r + \phi b^2 \quad (27)$$

$$\sigma_r = -p + \sigma_0(1-\eta\beta)\ln\left(\frac{r}{a}\right) + \frac{1}{2} \frac{\eta\beta}{(1-\nu^2)} \left[\frac{b^2}{a^2} - \frac{b^2}{r^2}\right]E\phi \quad (28)$$

$$\sigma_\theta = \sigma_r + \sigma_0(1+\eta\bar{\epsilon}^p) \quad (29)$$

where  $\sigma_0$ ,  $\eta$ ,  $\bar{\epsilon}^p$  are the initial yield stress, hardening parameter, and equivalent plastic strain, respectively, and

$$\bar{\epsilon}^p = \frac{-2}{\sqrt{3}} [\phi b^2/r^2 - (1-\nu^2)\sigma_0/E] / [1 + \frac{-2}{\sqrt{3}} (1-\nu^2)\eta\sigma_0/E] \quad (30)$$

$$\phi = [E\alpha_{22k} \frac{(c/b)^{2k} + 1}{(c/b)^{2k} - 1} - E\alpha_{12} + (1-2\nu)(1+\nu)]q/E \quad (31)$$

$$p = \sigma_0(1-\eta\beta)\ln \frac{b}{a} + q\left\{1 + \frac{1}{2} \eta\beta\left(\frac{b^2}{a^2} - 1\right)[Ak \frac{(c/b)^{2k} + 1}{(c/b)^{2k} - 1} + B + 1]\right\} \quad (32)$$

It is interesting to point out that  $p$  is a linear function of  $q$ . Similarly, when evaluating  $u$  at the bore from Eq. (27), we obtain

$$u_a/a = -(1-2\nu)(1+\nu)P/E + \phi b^2/a^2 \quad (33)$$

which can also be expressed as a linear function of  $q$  with the aid of Eqs. (31) and (32). Since the relation between  $q$  and  $u_b$  is linear from Eq. (11),  $p$  and  $u_a$ , given by Eqs. (32) and (33), respectively, can be expressed as linear functions of  $u_b$ .

#### FAILURE ANALYSIS

Since the steel liner is ductile and failure precedes by plastic flows, a nonlinear stress analysis beyond the elastic limit is required. The liner is considered as failure when the maximum stress or maximum strain reaches the ultimate limit ( $\sigma_u$  or  $\epsilon_u$ ). The steel is assumed to be elastic-plastic, linear strain-hardening with  $\sigma_0 = 120$  Ksi,  $E_t = 120$  Ksi, and  $\sigma_u$  (ultimate strength) = 140 Ksi. The composite jacket is elastically-orthotropic, and brittle failure is considered with the maximum strain criterion (ref. 5). The maximum strain from each simple test is either measured or computed from the measured strength divided by the elastic modulus. For the composite jacket used here, the maximum strain criterion is

$$-\epsilon_x'^* \leq \epsilon_\theta \leq \epsilon_x^* \quad \text{and} \quad -\epsilon_y'^* \leq \epsilon_r \leq \epsilon_y^* \quad (34)$$

where

$$\epsilon_x^* = X/E_\theta, \quad \epsilon_x'^* = X'/E_\theta, \quad \epsilon_y^* = Y/E_r, \quad \epsilon_y'^* = Y'/E_r \quad (35)$$

and  $X, X', Y, Y' = 262, 225, 8.7, 21.8$  Ksi, respectively.



## DISCUSSION OF RESULTS

Given any value of internal pressure, we can obtain numerical results for the stresses and strains in the radial and tangential directions and also for the displacement at any radial position in a compound tube. The actual specimens were constructed (ref. 1) using steel liners with two thicknesses and the appropriate thickness of the composite circumferentially wound on the liner. The geometric dimensions (a,b,c) for the three composite tubes are (0.9, 1.0, 1.189), (0.9, 1.07, 1.189), and (0.9, 1.07, 1.391) inch. The pressure at the interface between the liner and jacket has been obtained as a function of internal pressure and the results for three cases are shown in Figure 2. In this figure we also show the limits of internal pressure in the elastic-plastic range, i.e.,  $(p^*, p^{**}) = (20.48, 23.93)$ ,  $(23.06, 28.75)$ , and  $(27.47, 34.98)$  Ksi, respectively. The results of the hoop strains at the bore, interface between the liner and jacket, and outside surface for case 2 with (a,b,c) = (0.9, 1.07, 1.189) inch are shown in Figure 3 as functions of internal pressure. The complete (including elastic, elastic-plastic, and fully-plastic) ranges of loadings up to failure have been considered. The maximum value of internal pressure that this compound tube can contain without failure is  $p_f = 48.483$  Ksi, and the corresponding hoop strain is 1.12 percent. The results of hoop stresses at the bore ( $\sigma_\theta/a$ ) and at the interface ( $\sigma_\theta/b^-$  and  $\sigma_\theta/b^+$ ) are shown in Figure 4 as functions of internal pressure. It should be noted that the hoop stresses at the interface are discontinuous with  $b^-$  and  $b^+$  representing the location in the liner and jacket, respectively. Figure 4 shows very clearly that the results change drastically when yielding occurs. The relation changes from linear to nonlinear when yielding sets in and a more significant change occurs when the fully-plastic state is reached. The distribution of hoop stresses in the liner and jacket can be obtained at any given value of internal pressure. In Figure 5 we present the stress distributions for five values of internal pressure, i.e.,  $p = 23.065$ , 26.638, 28.751, 36.617, and 48.483 Ksi. The first three values correspond to initial yielding, 50 percent yielding, and 100 percent yielding. The percent yielding in the elastic-plastic range is defined by  $(p-a)/(b-a) \times 100$  percent. After the fully-plastic state is reached, the stress distribution changes drastically as shown in the figure for the last three values of internal pressure. The hoop stresses in the liner decrease slightly, but those in the jacket increase elastically as internal pressure is increased.

## CONCLUSION

The stresses, strains, and displacements in the liner and jacket can be obtained analytically for all loading ranges up to failure. The plastic deformation in the liner has a significant effect on the overall performance of the composite structure.

## ACKNOWLEDGMENTS

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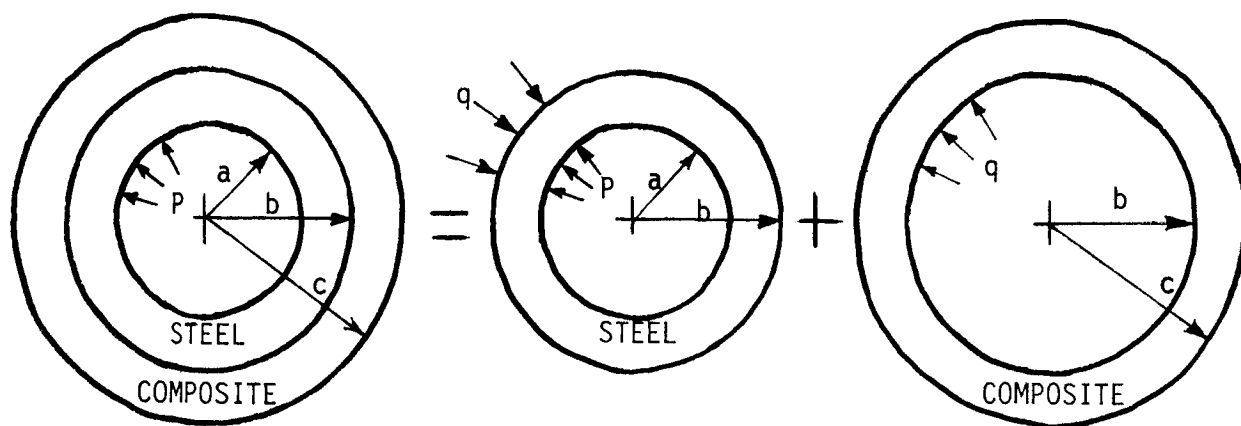


Figure 1 Schematic of a compound tube problem

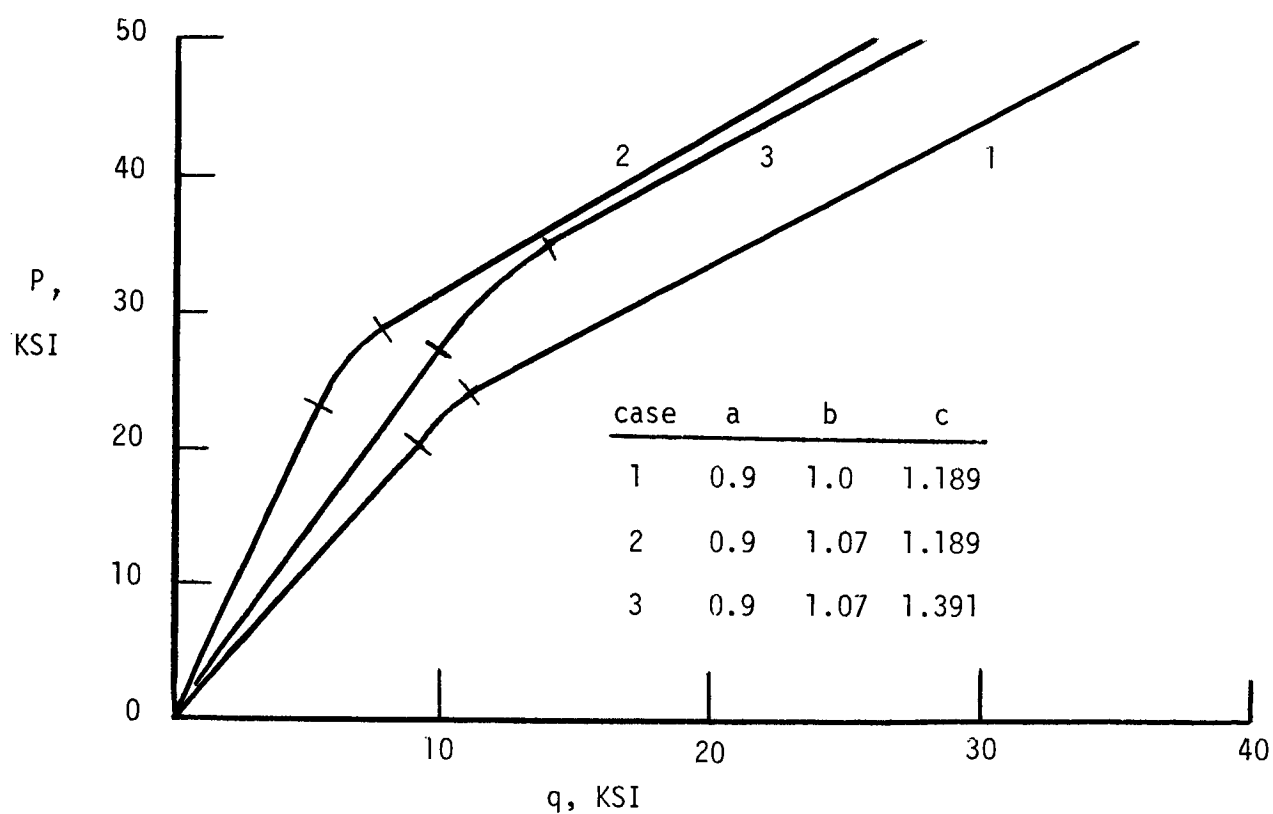


Figure 2 Interface pressure as a function of internal pressure

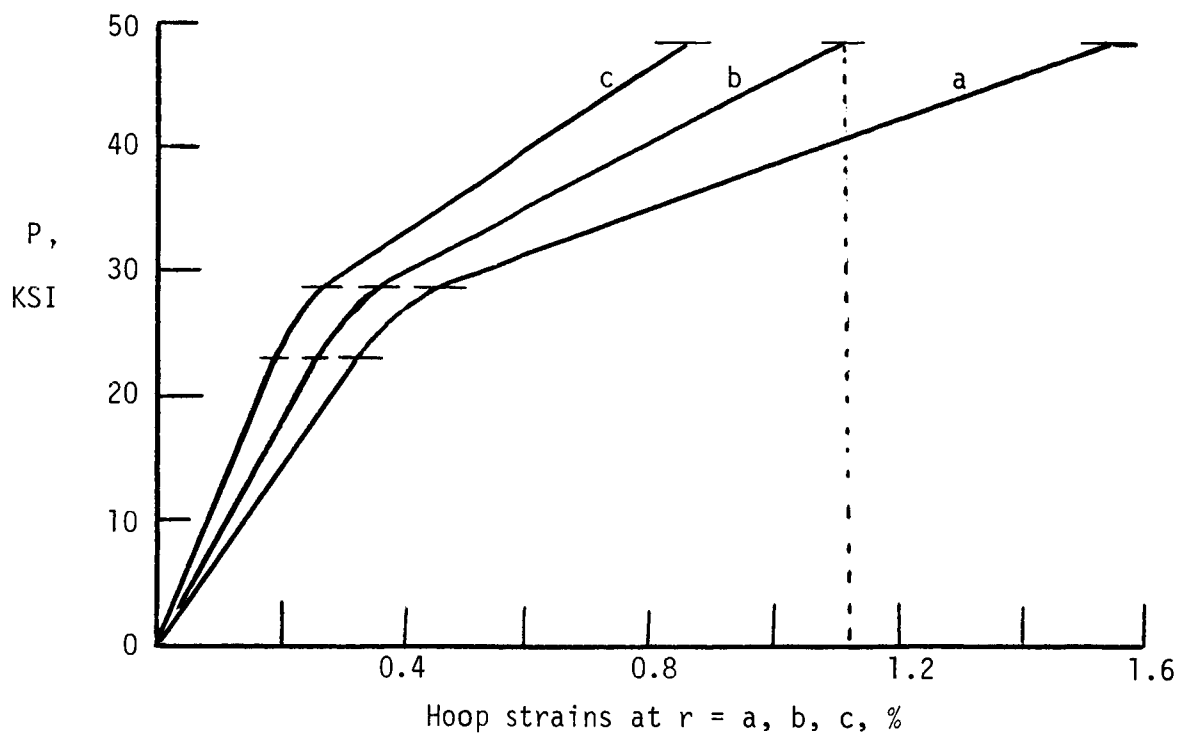


Figure 3 Hoop strains at the bore, interface and outside surface as functions of internal pressure.

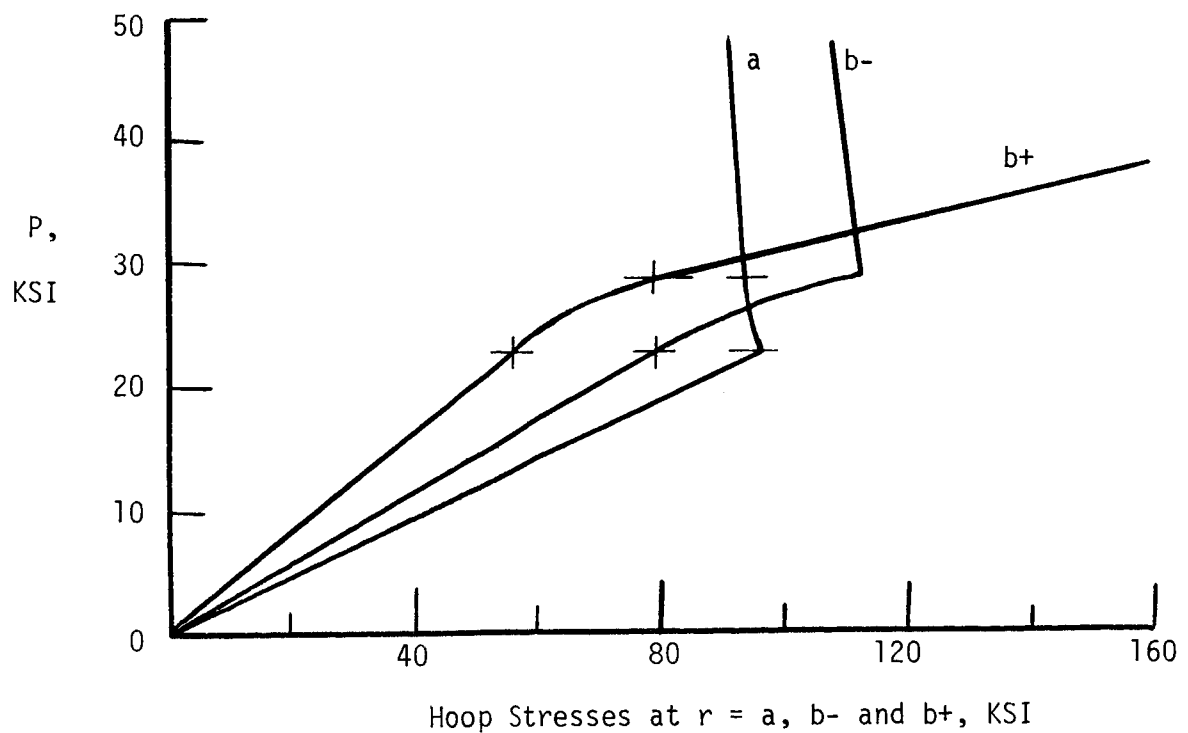


Figure 4 Hoop stresses at the bore and interface as functions of internal pressure

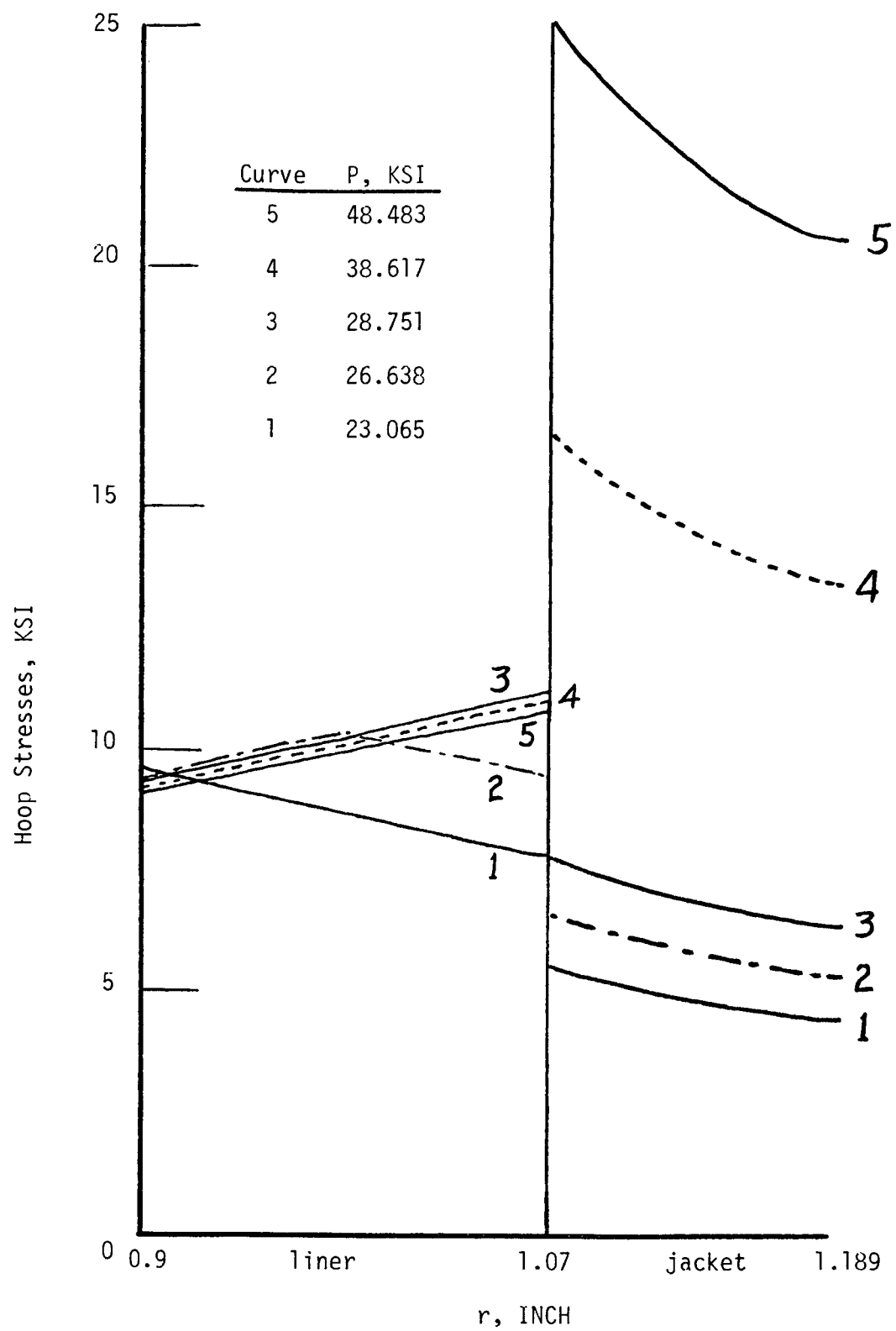


Figure 5 Distribution of hoop stresses in the liner and jacket

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10 83COMPOSITE FLIGHT-CONTROL ACTUATOR DEVELOPMENT<sup>1</sup>

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## SUMMARY

The composite actuator is "jam resistant", satisfying a survivability requirement for the Navy. Typically, the push-pull force needed to drive through the wound area of the composite actuator is 73 percent less than that of an all-metal actuator. In addition to improving the aircraft's combat survivability, significant weight savings were realized. The current design of the survivable, composite actuator cylinder is 36 percent lighter than that of the production steel cylinder, which equates to a 15 percent overall actuator weight savings.

## INTRODUCTION

Most actuators for flight critical control surfaces consist of two redundant hydraulic cylinders that are attached together and mechanically attached together in a dual-simplex or dual-tandem configuration. In conventional metal cylinder designs, deformation of one cylinder caused by ballistic impact will seize its piston. The second cylinder cannot develop the force required to push the seized piston past this deformation. Further, because the redundant actuators are fixed together mechanically, the undamaged cylinder jams as well.

The Naval Weapons Center, China Lake, California, contracted HR Textron Inc. (HR) in Valencia, California, to develop and produce a prototype, jam-resistant F/A-18 aileron actuator. Jam resistance is provided by an actuator cylinder constructed of filament-wound composite material. After ballistic impacts, the composite material shears away easily when the piston passes the deformed area. Actuators produced by HR under the Navy's contract were used for qualification, and for ballistic and flight testing.

A basic hydraulic actuator on the F/A-18 consists of two assemblies. The manifold assembly is a control unit that receives commands from the flight-control computers and provides

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<sup>1</sup>Performed under Naval Weapons Center Contract No. N60530-C-88-0260.

pressurized hydraulic fluid to the cylinder according to those signals. The cylinder assembly actually does the work by moving the control surface. Because development of the composite actuator was accomplished (1) on a low-risk basis and (2) intended mainly as a combat survivability improvement to reduce jamming, little developmental work was done on the manifold. The main areas of concern were components in the cylinder assembly.

The contract currently is nearing completion. The composite actuator for this development is in its final configuration. The cylinder assembly is composed of Amoco's T50 carbon fiber and Shell Epon 828 resin system, filament wound in a geodesic pattern over a thin 15-5PH steel liner. The liner acts as a durable wear surface for the piston seal, but does not produce heavy petalling when impacted ballistically.

Mounted on the cylinder is a production-level F/A-18 aileron manifold. To minimize structural discontinuities caused by porting through the composite cylinder walls, an interface plate between the manifold and cylinder allowed routing of fluid inlets and outlets via the ends of the cylinder.

Four complete actuator assemblies have been produced by HR under the Navy contract. Two were for qualification testing, to verify that the composite actuators would pass all tests required of the original production models. Qualification testing is brutal and essentially destructive; therefore, these two actuators will not be used for flight testing. However, they made acceptable articles for performing those ballistic tests that verified the jam-resistance concept. The ballistic testing was performed at the Naval Weapons Center.

The remaining two actuators are to be used for flight testing. Flight testing is to be conducted at the NASA Ames Dryden Flight Research Facility at Edwards Air Force Base, California, in September 1991. Flight testing is intended to demonstrate the maturity of the technology and to promote future development and application to production aircraft.

Although the actuators have been developed and produced as an "aircraft, combat survivability improvement," there are other benefits to the design. Foremost among these is the significant weight reduction realized through the use of composite materials. The composite cylinder in its current configuration achieved a 36 percent weight savings over existing production steel cylinders. This savings equates to a 15 percent actuator weight reduction overall. Actuators are a significant contributor to the weight of hydraulic systems; therefore, composite actuator construction provides an excellent opportunity for weight reduction in any military or commercial aircraft using hydraulic flight-control systems.

HR Textron has undertaken the cost of considerable IR&D to investigate additional areas of potential improvements. These

studies include the use of polyetheretherktone (PEEK) for end glands and fluid ports, and composite piston heads and push rods. With composites as a design feature at the start of a design cycle, there is little doubt that further weight reductions are possible.

#### GENERAL DESCRIPTION: FLIGHTWORTHY SURVIVABLE ACTUATOR

A direct replacement for the F/A-18 aileron servoactuator presents a difficult challenge because of the envelope in which the unit must fit. The current steel cylinder virtually fills the entire envelope. The production aileron servocylinder is a two-piece unit consisting of:

- An aluminum manifold that contains:
  - Electrohydraulic Servovalve
  - Solenoid Valve
  - Pressure Switch
  - Bypass Valve
  - Accumulator
  - Electrical Connector Interface
- A steel (15-5) cylinder that contains:
  - Monoball End Fittings
  - Piston Rod
  - Position Transducer
  - Cylinder End Closure (End Gland)
  - Associated Fluid Transfer Ports
  - Electrical Connector Interface
  - Cylinder-to-Manifold Bolting Provisions

The cylinder and some ancillary parts were redesigned, maintaining the identical interfaces with a manifold that was slightly modified for envelope reasons. Figure 1 shows the flight-verification, tested, and flight-test composite actuators.

Three major actuator components/areas were developed successfully with composite materials: cylinder, position-transducer, and mounting attachment. The following subsections describe the components/areas of the composite cylinder assembly.

#### Cylinder Design

The cylinder is a combination of filament-wound carbon fibers and unidirectional carbon fiber epoxy tape. Extensive evaluation -- to determine the exact location and orientation of each fiber -- is necessary to build a structure that is thick enough to withstand the system's pressure impulse cycles, yet thin enough to be within a reasonable envelope.

A liner is needed in the cylinder to prevent fluid permeation and provide a wear surface for the piston to ride upon. The liner must be selected judiciously so that strain and thermal coefficient compatibilities are met with the composite



overwrap structure. In addition, metal inserts with internal threads are necessary to engage the tailstock on one end and the end gland/retainer at the other.

The cylinder, liner and metal inserts are filament wound and cocured together to form an integral unit (Figure 2). The cylinder is fabricated on a mandrel. The surface finish of the mandrel is critical, to prevent scratches or gouges of the actual part during the extraction process, as well as to maintain dimensions of the part. The first fabrication step is to insert the liner and metal inserts onto the mandrel. Then, the winding process commences (i.e., alternate layers of filament wound fibers and longitudinal unidirectional tape are applied).

The whole unit, mandrel included, is cured in an oven. Curing temperature, of course, must be higher than the highest operational temperature of the unit. After the cure process, the unit is separated from the mandrel. The outside of the wound cylinder is rough in texture because of the winding process. A grinding operation can be performed to specific portions of the cylinder where other parts must be bonded to it. A final honing operation may be necessary to ensure conformance of the internal diameter dimensions of the cylinder.

#### Position Transducer

The position transducer used on the F/A-18 aileron is a dual-channel, self-checking type using 10 wires. These characteristics were left intact to maintain the original built-in testing (BIT) check and redundancy management concepts within the flight control computer. Null adjustment and rigging of the actuator remained identical to that of the original unit.

The only change made was for survivability reasons: to thin down the walls of the steel case (existing walls are 0.080 inch thick), and replace them with carbon/epoxy filament-wound material (Figure 3). This approach decreases the size of the petals formed when a projectile passes through, thereby creating less tendency for the fragment petals to jam the piston.

#### Manifold-to-Cylinder Attachment

The "mounting boss" is a design-critical area. This is because the attachment of manifold and cylinder assemblies depends on the boss. The barrel-mounting boss also is bonded secondarily (Figure 4) and fiber wrapped around the cylinder for support. The vibration spectrum at the aileron location in the F/A-18 aircraft is very severe; the mounting stresses between the cylinder and the manifold were the design driver.

## FLIGHT VERIFICATION TEST PROGRAM

The flight verification tests selected are critical to flight safety and operational requirements; they were completed in "Phase II" of the NWC program. A detailed review was conducted of the F/A-18 aileron Procurement Specification PS 74-690054 to identify critical tests for the composite actuator assembly and to qualify it as a flightworthy unit.

The actual flight-test time using the composite aileron actuator is to be a total of five hours. Thus, it seemed inappropriate to perform a complete spectrum of tests. The McDonnell Aircraft Company (MCAIR) aileron procurement specification (PS 74-690054), Paragraph 4.1.6 (Preflight Verifications), requires that only 10 percent of life cycling and impulse be performed prior to first flight. HR Textron suggested that time and funding resources would be saved if only 20 percent of the full test be performed; this suggestion was accepted by the Navy. In fact, after testing began, the composite servoactuators endured the full impulse cycling (ARP 1383), 40 percent life cycling and piston bottoming, and four low-temperature tests.

The tests (see Table 1) were conducted to demonstrate formally that the Hydraulic Aileron Composite Servoactuator Assembly is flightworthy and conforms to the Preflight Verification requirements of MCAIR specification PS 74-690054. Note that the tests described in Table 1, summarized for this paper, are but the highlights of a comprehensive flight verification/qualification program. A complete discussion of the actual test program (Ref. 1) appears to be beyond the intent of this conference.

Table 1. Critical Specifications for the Composite Actuator

| Test Title        | Objective                                                                                  | Conditions                                                        | Success Criteria                                                                                                                                   |
|-------------------|--------------------------------------------------------------------------------------------|-------------------------------------------------------------------|----------------------------------------------------------------------------------------------------------------------------------------------------|
| Life Cycling      | Verify wear life without failure                                                           | 200,000 cycles at 225°F fluid temperature                         | 1) Dynamic seal of 1.013M cycles min.<br>2) No deterioration of performance.<br>3) No cracks or excessive wear.                                    |
| Piston Bottoming  | Verify actuator piston snubber is operating satisfactorily with impact load within limits. | 13,000 cycles at 170°F fluid temperature.                         | 1) Peak axial loads shall not exceed 3023 pounds.<br>2) There shall be no structural failure of the unit.<br>3) Disassembly inspection for damage. |
| Impulse Cycling   | Verify fatigue strength of cylinder under pressure impulse conditions.                     | 40,000 cycles at 230-4500-230 psi, 40,000 cycles at 0-2250-0 psi. | 1) No external static seal leakage.<br>2) No deformation or structural failure.                                                                    |
| Low Temperature   | Verify satisfactory performance at -40°F.                                                  | -40°F ambient and fluid temperature                               | 1) No excessive leakage.<br>2) No permanent performance deterioration.                                                                             |
| High Temperature  | Verify satisfactory performance at 275°F.                                                  | 240°F ambient temp., 275°F fluid temp.                            | 1) No excessive leakage.<br>2) No permanent performance deterioration.                                                                             |
| Temperature Shock | Verify ability of actuator to withstand extreme temperature variations.                    | -40° to 240°F ambient temp., -40°F to 275°F fluid temp.           | No binding or permanent damage during operation.                                                                                                   |

|                |                                                   |                                               |                                                                                                                                   |
|----------------|---------------------------------------------------|-----------------------------------------------|-----------------------------------------------------------------------------------------------------------------------------------|
| Vibration      | Verify structural integrity under vibration.      | Production F/A-18 aileron vibration spectrum. | 1) No binding or excessive external leakage.<br>2) No cracks or loose parts.<br>3) Actuator shall work satisfactorily after test. |
| Proof Pressure | Verify actuator to withstand proof pressure.      | 4500 psi at 275°F ambient and fluid temp.     | No deterioration or damage.                                                                                                       |
| Burst Pressure | Verify safety under hydraulic overload condition. | 7500 psi                                      | Withstand burst pressure without rupture.                                                                                         |

#### Test Environment

The standard test conditions during the program were as stated below:

- Supply Pressure: 3000 ± 100 psig
- Return Pressure: 235 psig ± 50
- Ambient Temperature: 55°F to 95°F
- Fluid Temperature: 70°F to 140°F
- Test Fluid: MIL-H-83282 in hydraulic system within fluid cleanliness limits of NAS 1638 Class 6
- Barometric Pressure: Local Ambient
- Relative Humidity: 20% to 90%

Test parameter tolerances were as follows:

- Temperature: ±2.5°F
- Barometric Pressure: ±5%
- Relative Humidity: ±5%
- Current: ±1% Full Scale
- Voltage: ±1% Full Scale
- Hydraulic Pressure: ±1% Full Scale
- Acceleration: ±10%
- Force: ±5%
- Position: ±0.010 inch

Some of the highlighted tests are illustrated in the following figures:

- Figure 5 - Life Cycling
- Figure 6 - Piston Bottoming
- Figure 7 - Impulse Cycling
- Figure 8 - Low Temperature

- Figure 9 - High Temperature and Temperature Shock
- Figure 10 - Vibration

### Test Results

The flightworthiness testing was accomplished on two units. One qualification unit is used for structural testing (i.e., life and impulse cycling, piston bottoming, etc.); the other unit is used for environmental testing (i.e., temperatures, vibration, etc.). Note that the units did experience some "failures" during the test program. However, the failures were not associated with the composite structures and were remedied easily. For example, failures included electrical failures within the manifold and the fatigue failure of a metallic tube; they were corrected quickly.

The composite servoactuators passed successfully and, in some cases, exceeded program requirements. The structural qualification unit passed all the tests required, plus the following additions:

- Five Acceptance Test Procedures<sup>2</sup> (ATPs)
- 400,000 Life Cycles (200,000 cycles were required)
- 26,000 Piston Bottoming cycles (13,000 cycles were required)
- 200,000 High-Pressure Impulse Cycles (40,000 cycles were required)
- 200,000 Low-Pressure Impulse Cycles (40,000 cycles were required)

The environmental qualification unit passed all the tests required, plus the following additions:

- Four Acceptance Test Procedures (ATPs)
- Four low-temperature tests (one was required)

### Ballistic Tolerance

As discussed earlier, ballistic tests were conducted on the F/A-18 aileron composite actuators. The Naval Weapons Center (NWC) test facility and its setup for the ballistic tests are shown in Figure 11. The tests employed different threats at specific impact velocities and various locations. Results indicated that the composite cylinders/actuators required

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<sup>2</sup>An ATP consists of physical inspection for defects, proof pressure, insulation resistance, dielectric strength, operation, external and internal leakages, main ram friction, output travel, main ram LVDT, output ram velocity, frequency response, failure transient, damper operation, chatter and instability, performance duty cycle, and seven more pressure and electrical related tests.

significantly lower forces, (Fig. 12) to "drive through" the damaged areas than the all-metal version.

The Navy's tests verified the HR approach for flight control actuator design. The basic characteristics that an actuator must possess were demonstrated in the composite actuator: wear quality, fluid impermeability, endurance, thermal stability, and strength within the realm of the F/A-18 aileron actuator specifications. Jam-resistance of the composite actuator was substantiated. The replacement of metal actuator components (cylinder, position transducer housing and mounting attachments) with carbon epoxy materials is a reasonable option and has low design risks.

### SUMMARY AND CONCLUSIONS

Significant accomplishments have been made in identifying the advanced materials technologies best suited to the flight control actuators. Some of these areas are discussed briefly in this paper. Other advanced material components -- such as thermoplastic end glands and fluid ports, and composite piston heads -- have shown promising results in tests (Ref. 2). The transition of composite technology to actuation technology is here. Different composite actuator designs are presently undergoing qualification testing; however, there is a need to extend the research and development for general applications.

A relatively large amount of metal components (at least 41 percent by weight) still exists in current composite actuator designs. The metal components have not yet been replaced by advanced materials because of high loads and tight envelope requirements. R&D activities are underway to look at these high-risk actuator components and develop new processes such as thermoplastic filament winding and molding techniques, and application of new materials (thermoplastics and metal-matrix composites).

In addition to the continuing investigation of new processes and materials, we recognize that fabrication and material costs must be competitive. The demands for fibers and resins are increasing; processes such as filament winding and injecting molding are combining to bring acquisition and support costs down.

Ongoing advanced development efforts will reduce the weight further and potentially, the cost. Also, these developments will increase the survivability of flight control components. Efforts in this challenging technology will help meet the demands of the 1990s and will contribute to the success of early twenty-first century vehicles.

#### REFERENCES

1. Ching, F., "Jam-Resistant Flight Control Actuator (F/A-18 Aileron) Development Program," NAWC WPNS TP 8009, March 1992.
2. Ching, F., "Advanced Materials Flight Control Actuators," SAE 901048, Apr. 1990.

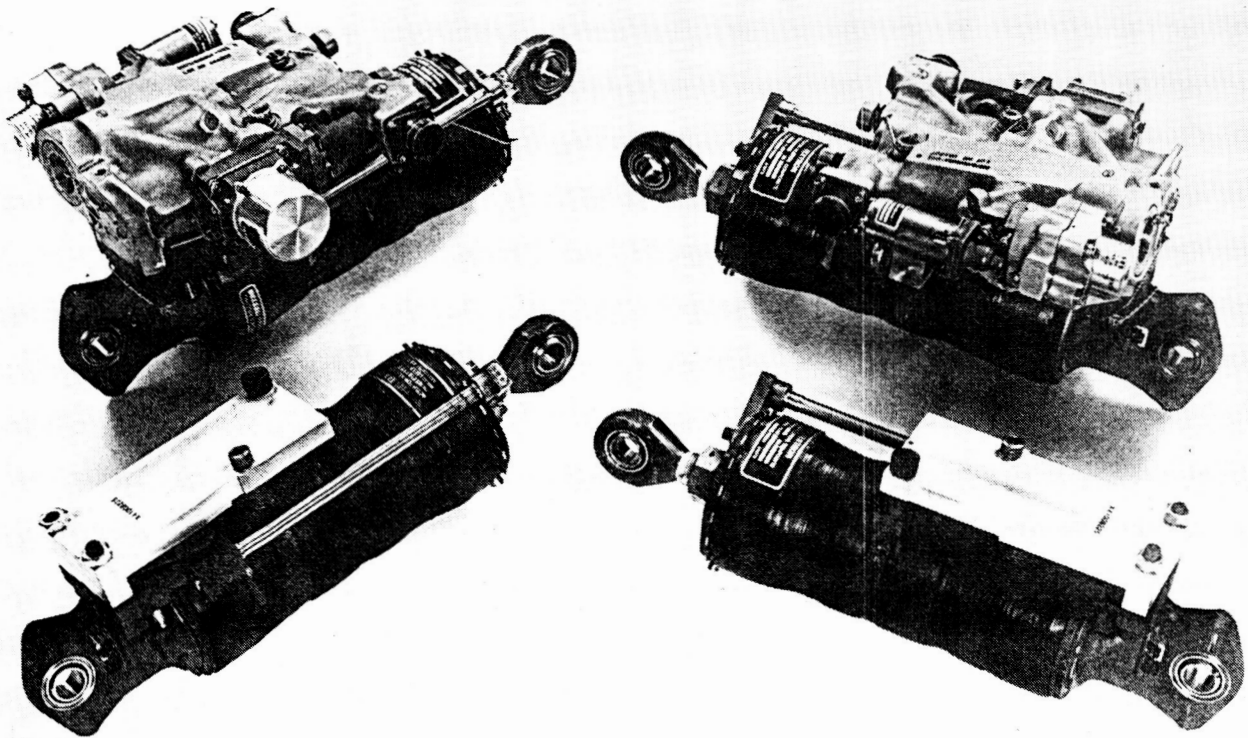


Figure 1. Jam-Resistant Flight Control Actuators (F/A-18 Aileron)

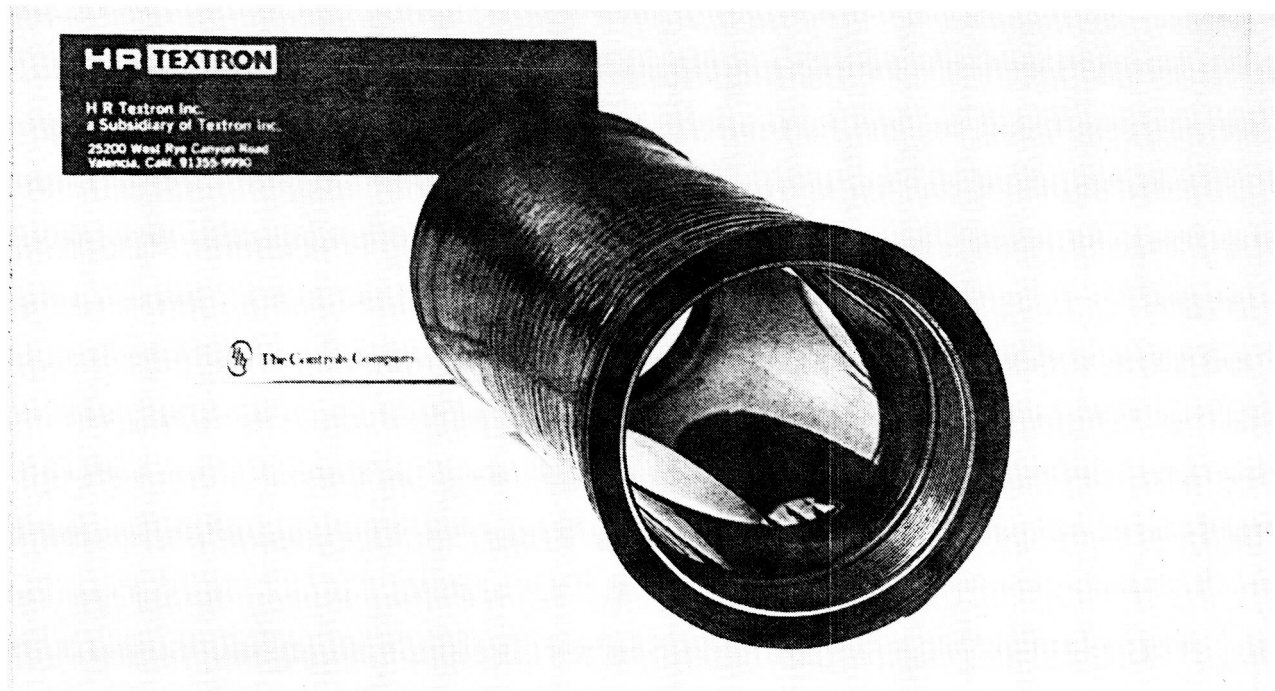


Figure 2. Carbon Epoxy Cylinder with Steel Liner and Thread Inserts

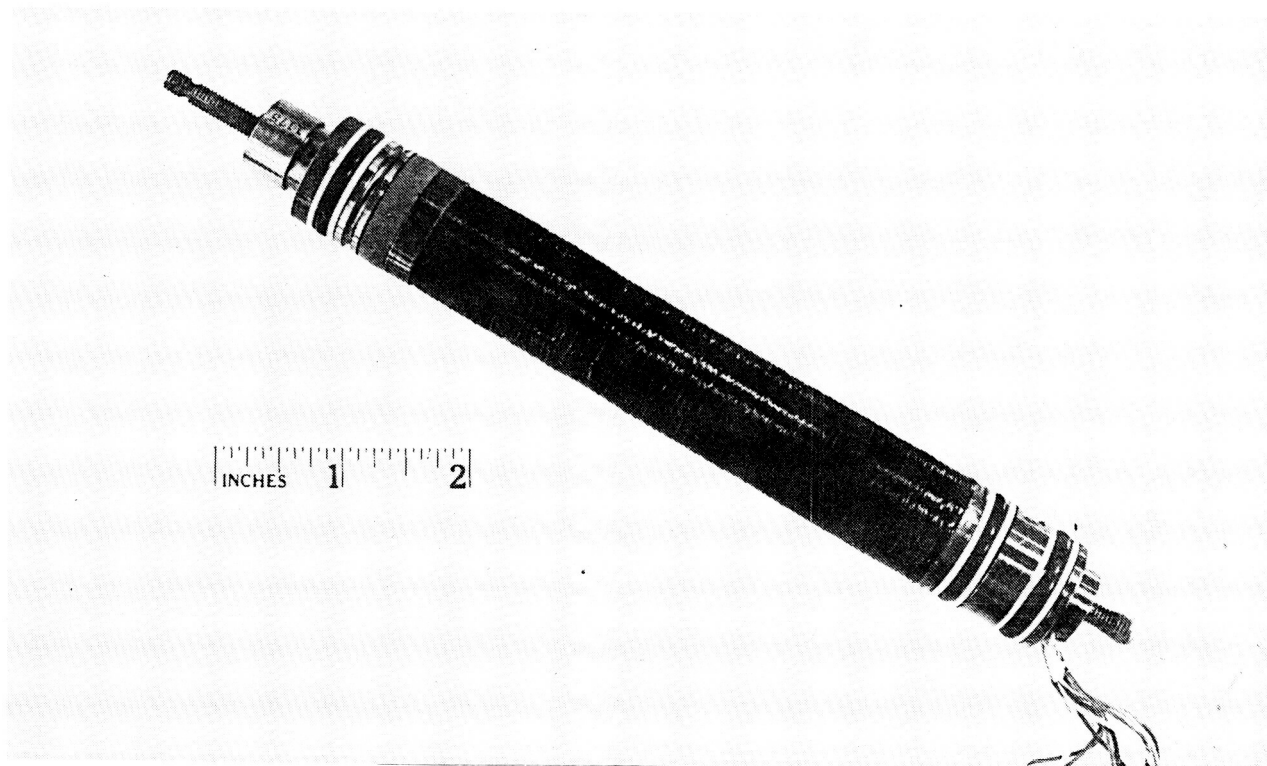


Figure 3. Composite-Wrapped LVDT



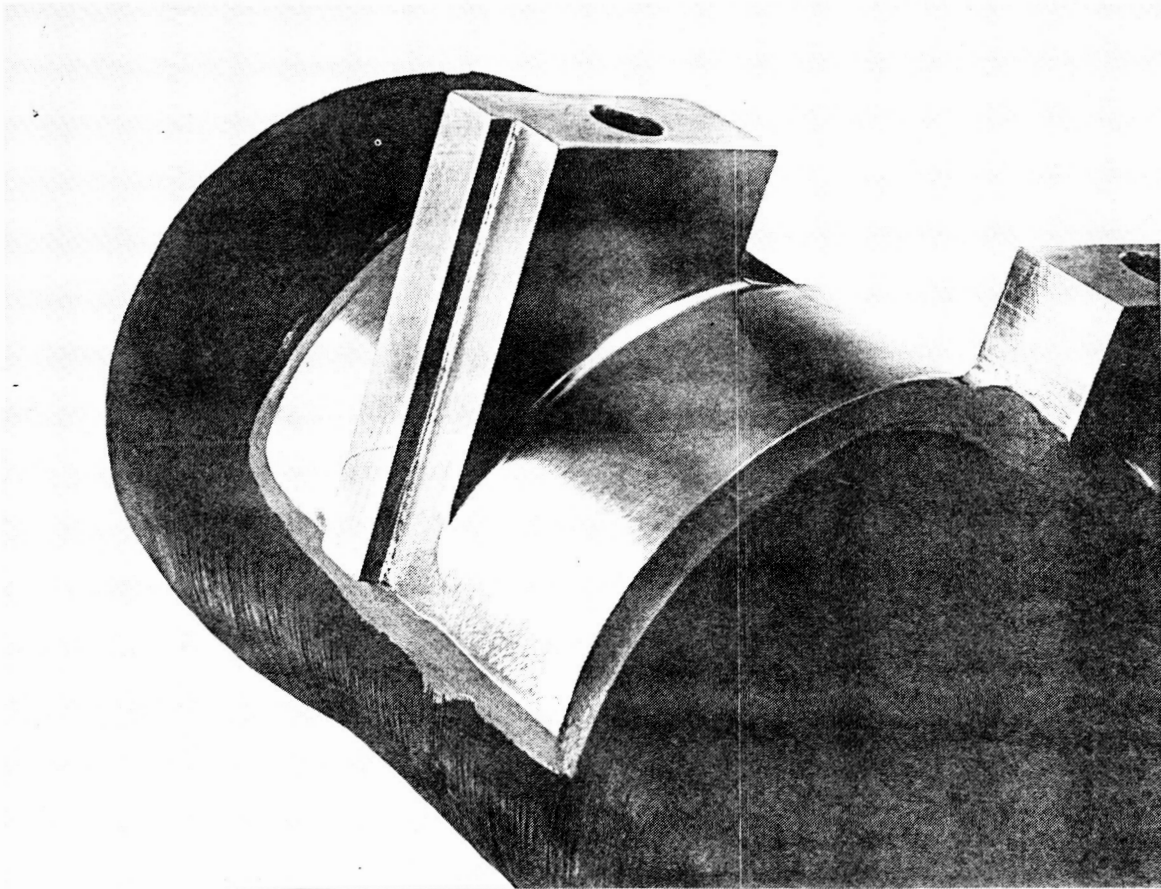


Figure 4. Bonded Mounting Boss (not wrapped)

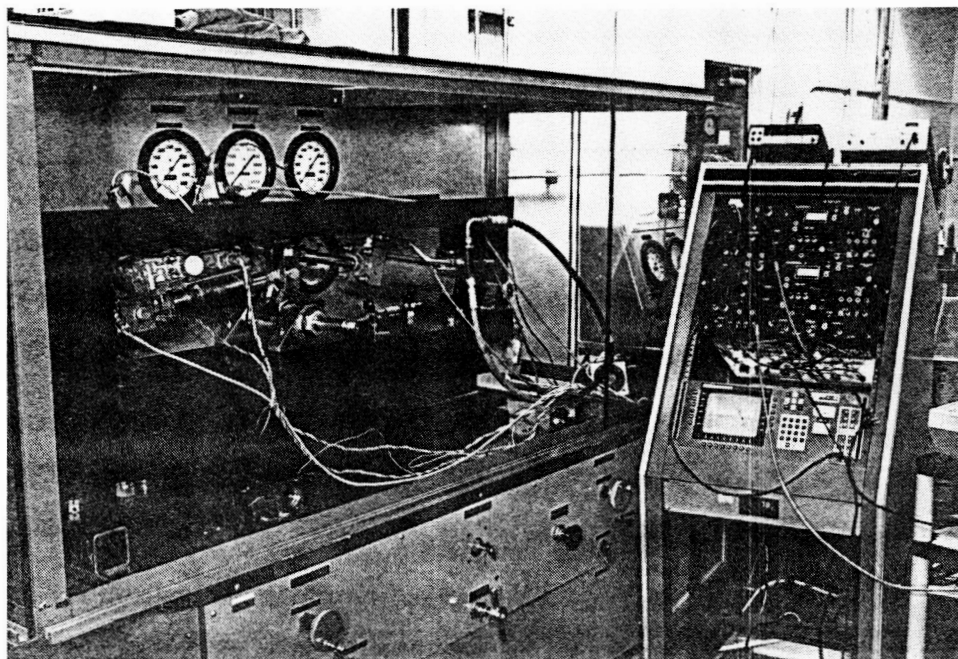


Figure 5. Life-Cycling Fixture and Instrumentation

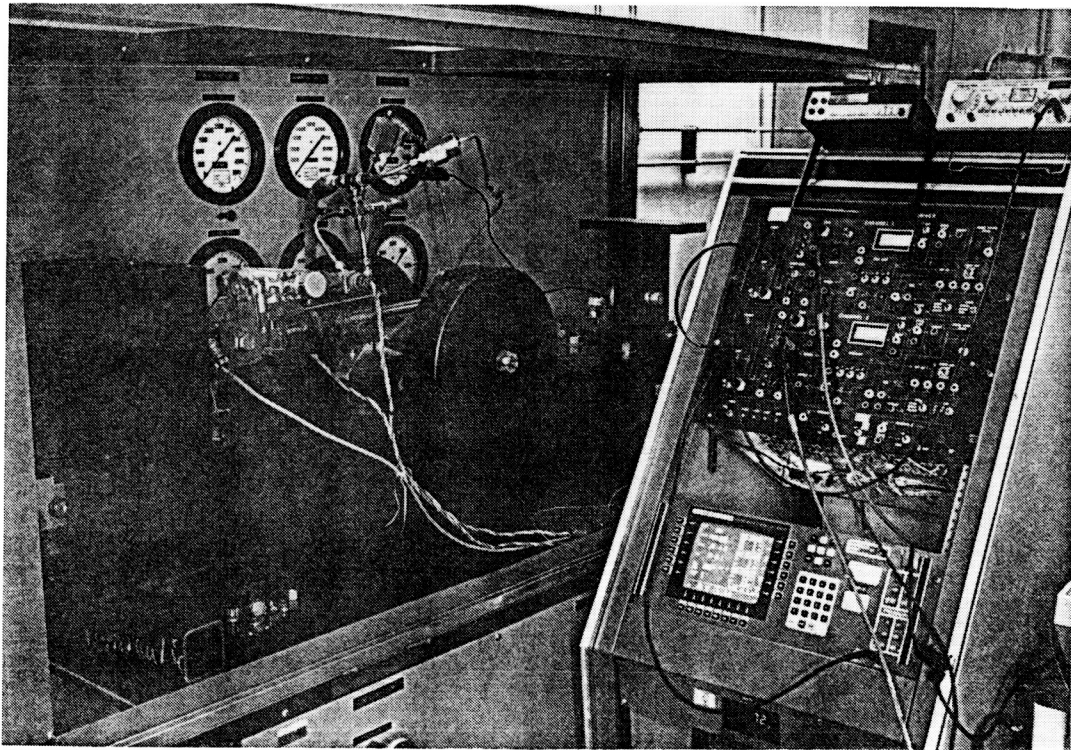


Figure 6. Piston Bottoming Setup

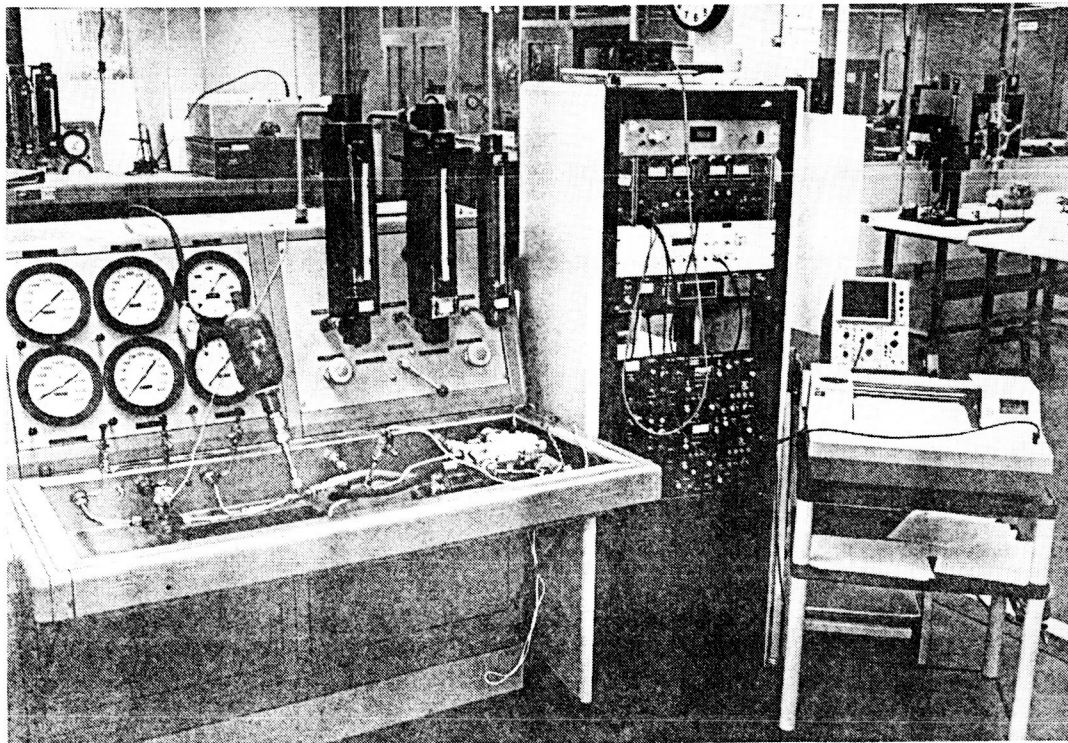


Figure 7. Impulse Setup and Instrumentation



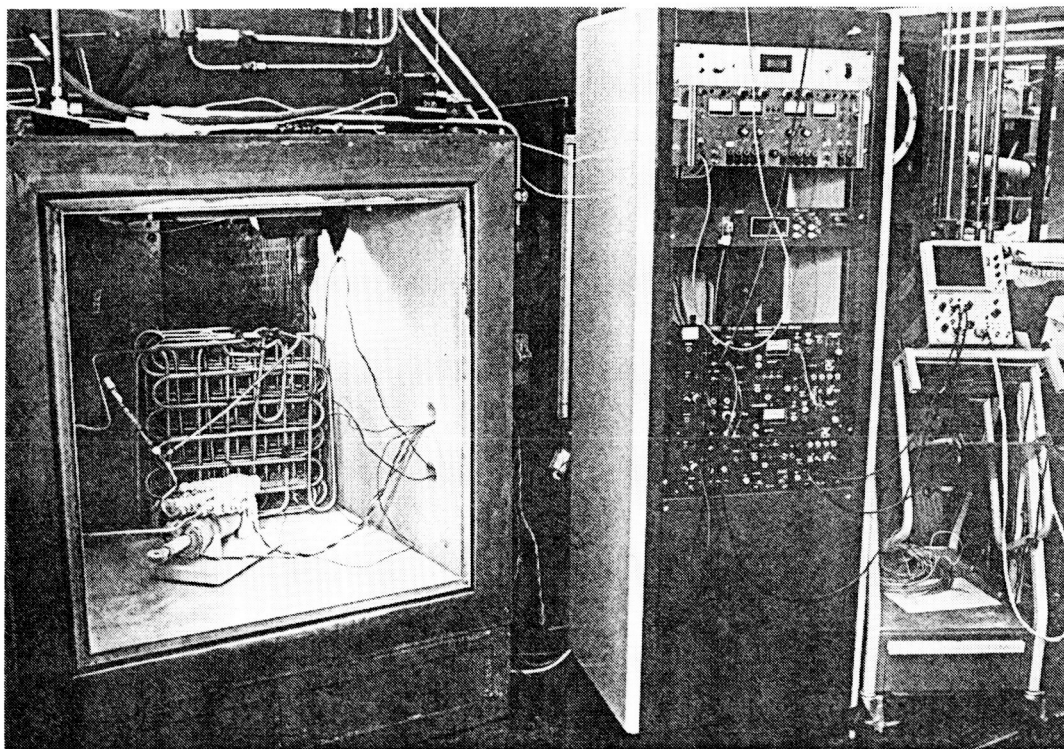


Figure 8. Low Temperature Setup

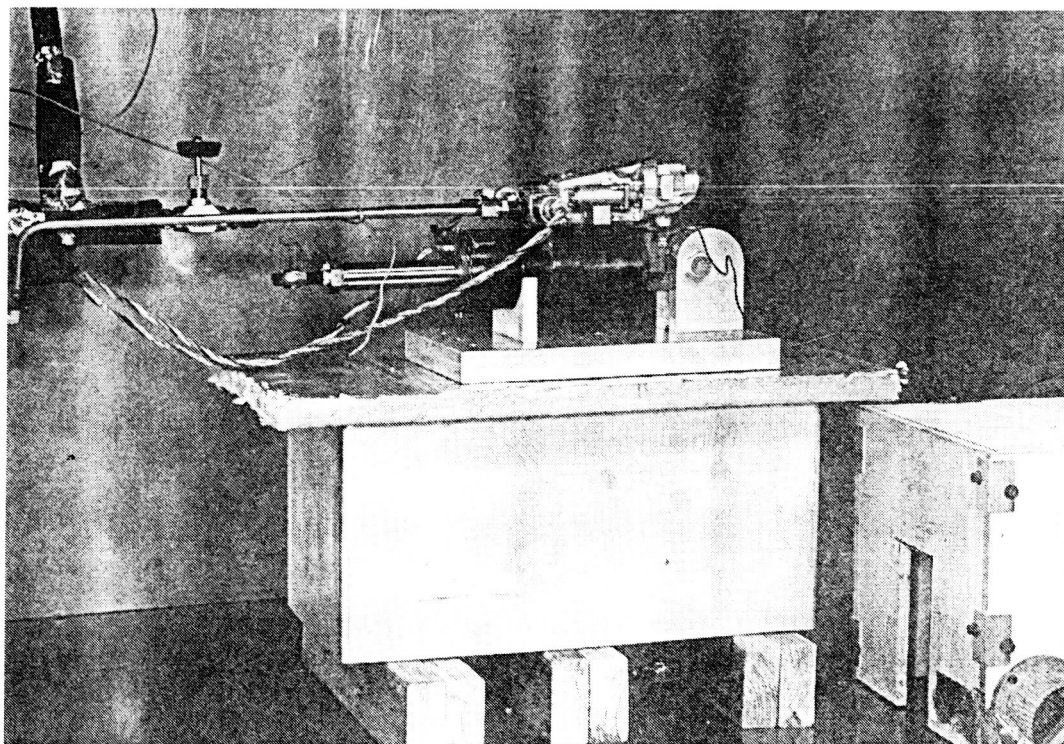


Figure 9. High Temperature/Temperature Shock Setup

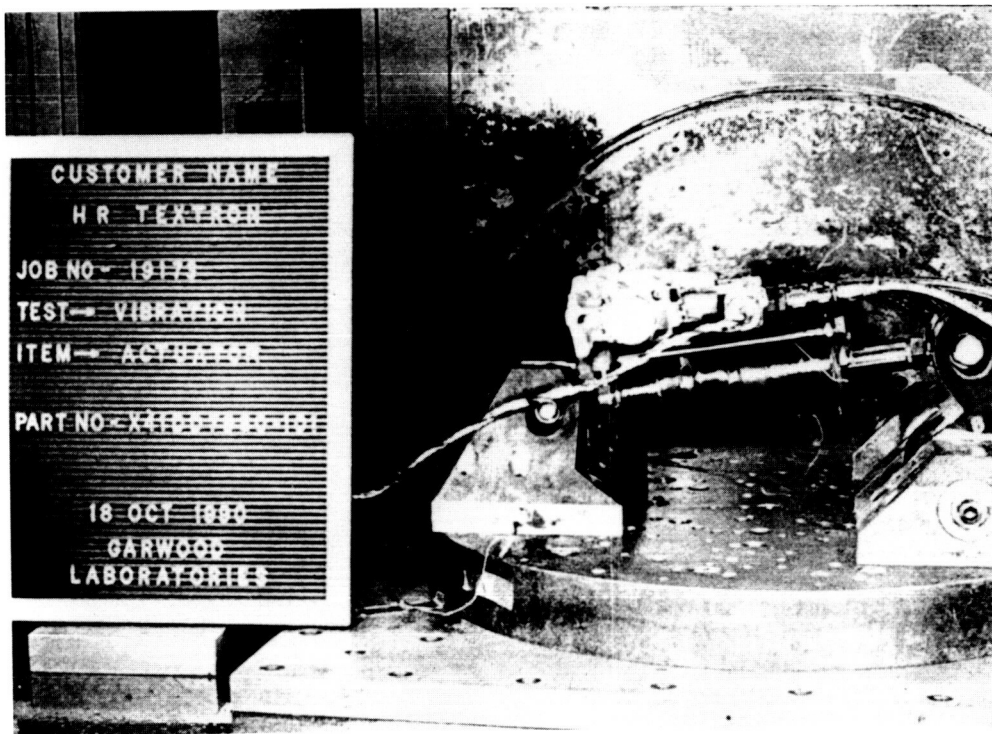


Figure 10. X-Axis Vibration Setup

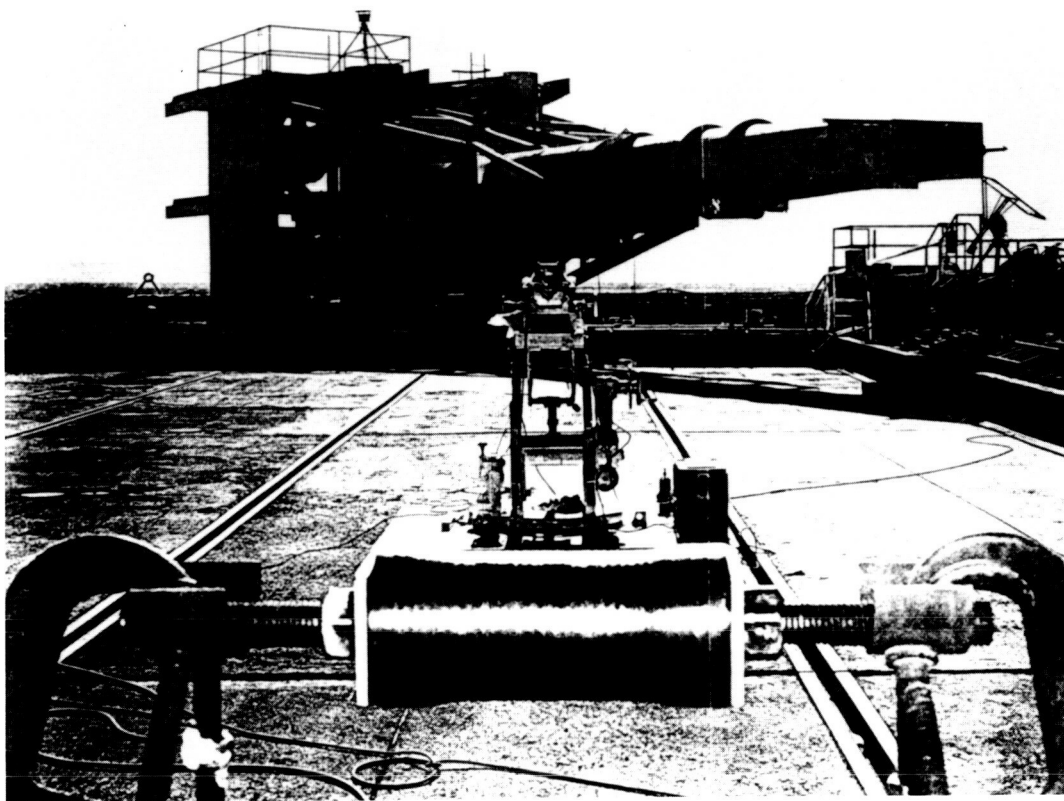


Figure 11. NWC Ballistic Test Facility

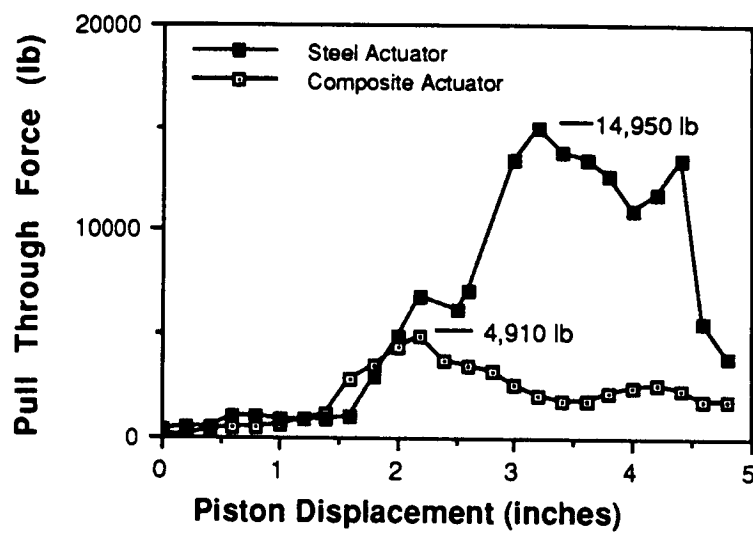


Figure 12. Ballistic Test Comparison

**SESSION IX**  
**THICK STRUCTURES TECHNOLOGY**

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HIGHER ORDER FINITE ELEMENT ANALYSIS OF THICK COMPOSITE LAMINATES<sup>1</sup>

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## SUMMARY

A higher order, sub-parametric, laminated, 3-D solid finite element has been used for the analysis of very thick laminated composite plates. The geometry of this element is defined by four nodes in the X-Y plane which define a prism of material through the thickness of the laminate. There are twenty-four degrees of freedom at each node; translations at the upper and lower surfaces of the laminate in each of the three coordinate directions, and the derivatives of these translations with respect to each coordinate. This choice of degrees of freedom leads to displacement and strain compatibility at the corners. Stacking sequence effects are accounted for by explicitly integrating the strain energy density through the thickness of the element.

The laminated solid element has been combined with a gap-contact element to analyze thick laminated composite lugs loaded through flexible pins. The resulting model accounts for pin bending effects that produce non-uniform bearing stresses through the thickness of the lug. A thick composite lug experimental test program was performed, and provided data that was used to validate the analytical model. Two lug geometries and three stacking sequences were tested.

## INTRODUCTION

Composite lugs provide a mechanism for the transfer of concentrated loads from one structural member to another. The most notable examples are lugs that transmit wing loads into carry-through bulkheads. The geometry of these lugs can vary substantially for different applications, but they are typically very thick ( $\gg 1$  inch) and may be required to carry out-of-plane as well as in-plane loads. In addition, effects such as pin bending may result in complex stress states through the thickness of the laminate, even when it is only loaded in-plane.

The wide range of variables associated with the design and analysis of thick composite lugs necessitates the use of an analytical model that is adaptable, both in the range of geometries and types of load conditions that can be analyzed. The complex stress distributions that may develop through the thickness require that the model be capable of predicting

<sup>1</sup>The work described in this paper was performed under the NASA ACT contract NAS1-18862, 'Innovative Composite Aircraft Primary Structures.'



three dimensional stress fields. The finite element method is one such method, and is the method that was chosen for this program.

Aircraft components that are fabricated from laminated composites are typically analyzed using plate elements that are based on classical laminated plate theory. These elements are simple to use, since the geometry of the element can be defined in two dimensions, and they account for in-plane and out-of-plane loads. They are of limited use for the analysis of thick lugs, however, since their derivation permits only linear variations in the in-plane stresses through the thickness of the element, and out-of-plane stresses are assumed to be negligible.

At the other extreme, a thick lug can be modeled with three dimensional solid finite elements. Using this approach, each ply can be modeled discretely with one or more elements through the thickness of each ply, or groups of plies can be lumped into a single element. These elements are based on assumed three dimensional displacement fields, and allow complete generality in defining the lug geometry and loads. Although models generated with these elements provide accurate results, their use is cumbersome and they require substantial computing resources. They are therefore not recommended for the type of parametric study that would be required to optimize a lug design. The approach described in this paper is based on a subparametric laminated solid element and represents a compromise between the two methods described above.

#### ANALYTICAL DEVELOPMENT

The thick composite analysis developed for this program uses a subparametric laminated solid element that was developed in [1]. Geometric shape functions for the element are linear in the x-y plane and constant in the z direction, and displacement fields are defined by cubic shape functions in three dimensions. The element geometry is defined by four nodes that specify the (x,y) positions of the edges of a right prism (Figure 1), and displacement degrees of freedom are retained at each of the eight corners of this prism. Twelve degrees of freedom are specified at each corner; the translation in each coordinate direction (3 degrees of freedom), and the partial derivatives of each translation with respect to each coordinate (9 degrees of freedom).

Like conventional homogeneous solid finite elements, the stiffness matrix for the laminated solid element is generated by integrating the strain energy density over the volume of the element. However, the effects of the stacking sequence are included in the laminated solid element by performing the integration numerically over the volume of each ply contained in the element, and summing the result. This process also provides much of the information that will be required in subsequent calculations to determine the strains in each ply.

To account for pin bending effects, both the lug and the pin must be modeled. The generation of these models is relatively simple, since they are explicitly defined in only two dimensions. The through-the-thickness geometry is defined by specifying the ply thickness and stacking

sequence. A typical model is shown in Figure 2. The lug/pin contact is modeled by gap/contact springs that couple lug and pin displacements in the radial direction. The gap/contact springs are bi-linear elements that have a very high stiffness when the gap is closed (the spring is in compression), but zero stiffness when the gap is open (the spring is in tension). Since the extent of the contact area is not known a priori, the analysis must be performed iteratively to determine which gap/contact elements are closed and which are open. The iterative analysis is simplified by treating the lug and pin as substructures which have been reduced such that only degrees of freedom connecting to gap/contact elements are retained. The iterative portion of the analysis is then performed with a model that has only a few dozen degrees of freedom.

The converged solution from the iterative portion of the analysis is used to back substitute for the displacement solution throughout the lug and pin models. The displacement solution is then used to perform a laminate analysis on an element by element basis. For each element, the average strain in a given ply is calculated by integrating the strain field over the volume of that ply in the element, and the average strain in an interface is calculated by integrating over the interface area, as shown in Figure 3. These strains are then used to calculate average stresses in each ply and interface, which are used to perform a failure analysis.

The failure analysis considers three types of failure; 1 - matrix failure within a ply, 2 - fiber failure within a ply, and 3 - failure of an interface. The failure criteria used to assess the failure of a ply is based on Hashin's criteria for a 2-D laminate [2], extended to account for all six stress components. Interfaces are assumed to be isotropic matrix rich regions and failure is assumed to occur when the von Mises stress exceeds the shear allowable for the matrix. Ultimate failure of the laminate is assumed to occur when a fiber failure is predicted for a ply.

## EXPERIMENTAL STUDY

A test program was conducted to investigate the effects of stacking sequence on the strength of thick lugs. The major goal of this study was to determine how stiffness should be distributed through-the-thickness of a laminate to optimize strength. All of the lugs were fabricated from AS4/APC-2, and had the same thickness and external geometry. Combinations of 0,  $\pm 45$ , and 90 degree plies were stacked into four different 30 ply sublaminates, and consolidated in a hydraulic press. Six sublaminates (two each of three different stacking sequences) were combined to form three different 180 ply laminates and co-consolidated in an autoclave. A summary of the lugs tested in this program is shown in Figure 4.

Each laminate contained a similar number of plies in each direction, but their distribution through the thickness was different. This approach produced laminates with equal in-plane properties, but different through-the-thickness stiffness distributions. Two pin sizes were used to investigate different failure modes in the lugs. A 1.00 inch diameter

steel pin was used to generate bearing failures, and a 1.75 inch diameter steel pin was used to generate net section failures. Axial loads were applied to the lugs through the pins by means of a clevis that allowed a .1 inch gap on either side of the lug (Figure 5). A constant gap size (0.1") was maintained throughout the test.

All testing was performed at room temperature with a load rate of 500 lb./sec., and all lugs were tested in as received moisture condition. As expected, the lugs with 1.75 inch diameter holes exhibited a catastrophic fiber failure of the net section, and the lugs with 1.0 inch diameter holes showed permanent yielding around the hole prior to experiencing shear/bearing failures. The initial bearing failure load was determined by observing the behavior of axial strain data from rosettes located 0.5 inch away from the edge of the 1.0 inch hole and 0.25 inch away from the 1.75 inch hole at the center line of specimens. Load-strain data collected during the tests indicated axial strain decreases associated with bearing failure ahead of the 1.0" pin, as shown in Figure 6.

## RESULTS AND DISCUSSION

Models of the test lugs were generated using the laminated solid element. All models used the 2-D mesh shown in Figure 7, which took advantage of the symmetry of the lug to reduce the size of the analytical problem. A convergence study was performed to determine if the through-the-thickness stress distribution in the lug could be predicted using only one element through the thickness. This study showed that the through-the-thickness stress distributions were very complex, and that multiple elements were required to approximate the gradients near the surface of the lug. The results of this study are shown in Figure 8. Four elements were used through the thickness of all subsequent models.

Analysis of the lugs indicated that, for bearing critical lugs (1.00 inch diameter pin), the stacking sequence which placed stiffer material at the center of the lug would be the strongest, and for net section critical lugs (1.75 inch diameter pin), the lug with uniformly distributed stiffness would be strongest. This analytical prediction was confirmed by static tests performed on the lugs. However, the magnitude of the differences in strengths of the various stacking sequences was not as significant as had been predicted. A summary of the analytical and experimental results is shown in Figure 9.

All analytically predicted strengths were within 8% of the experimental values. The largest differences were for the stacking sequences that placed stiffer material near the surfaces of the lug. The analytical models used elements that were of nearly equal thickness, and it may be possible to improve the analytical results by placing fewer plies in the elements near the surface of the lug where the stress gradients are the highest.

In general, the subparametric laminated solid element provides a reasonably accurate means for predicting the complex state of stress in thick laminates. A computer program, based on the use of this element,

has been developed at MCAIR. This program allows an analyst to define the geometry of a thick laminate in only two dimensions, even when there is more than one element through the thickness. Although they were not important to the studies performed in this program, the laminated solid element includes out-of-plane degrees of freedom, which will allow the element to be used to study lugs that are loaded both in and out of plane.

#### CONCLUDING REMARKS

The objective of this study was pursued through parallel efforts in finite element model development and structural testing of thick thermoplastic composite lugs. The analysis of these lugs in conjunction with the experimental results support the following conclusions.

- An accurate means of predicting the complex state of stress in thick composites has been demonstrated through the use of the higher order finite element formulation.
- Caution should be used in the modeling and analysis of thick composite sections since multiple elements may be required through-the-thickness even for higher order elements, as demonstrated in the convergence study.
- Accuracy in predicting the pin bending effect is unclear. The difficiencies in failure loads for the different stacking sequences tested were not large enough to make an accurate assessment of pin bending effects possible.

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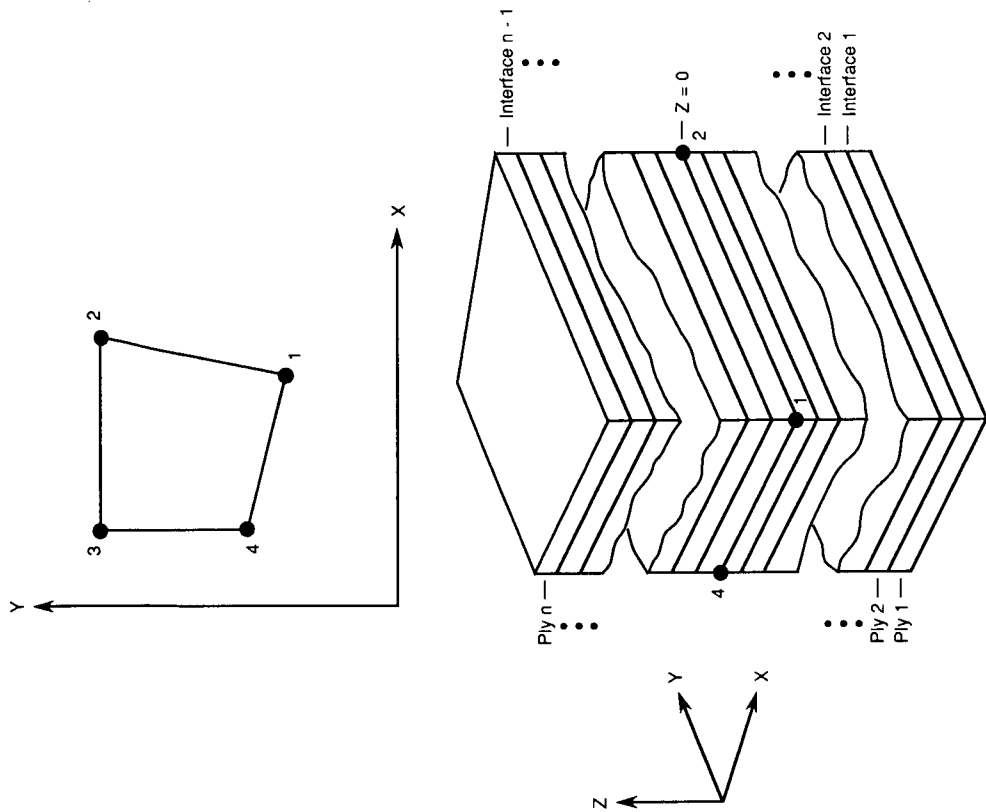
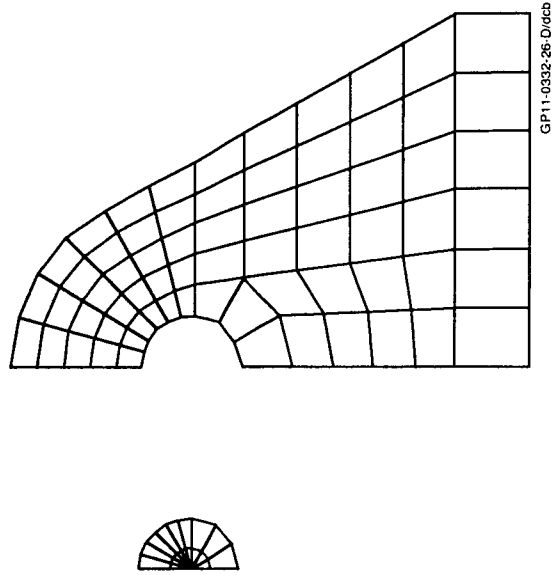


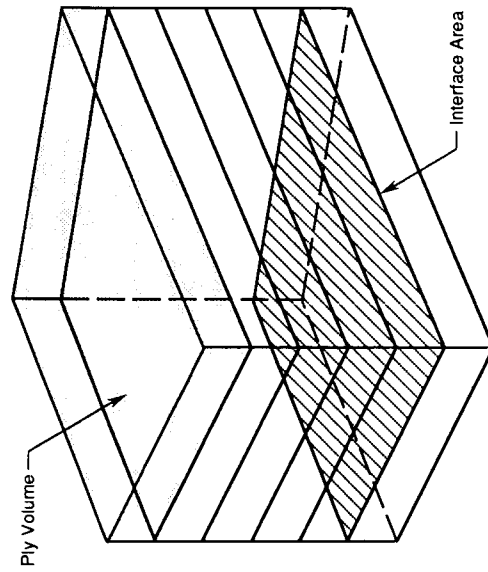
Figure 1. Subparametric Laminated Solid Element

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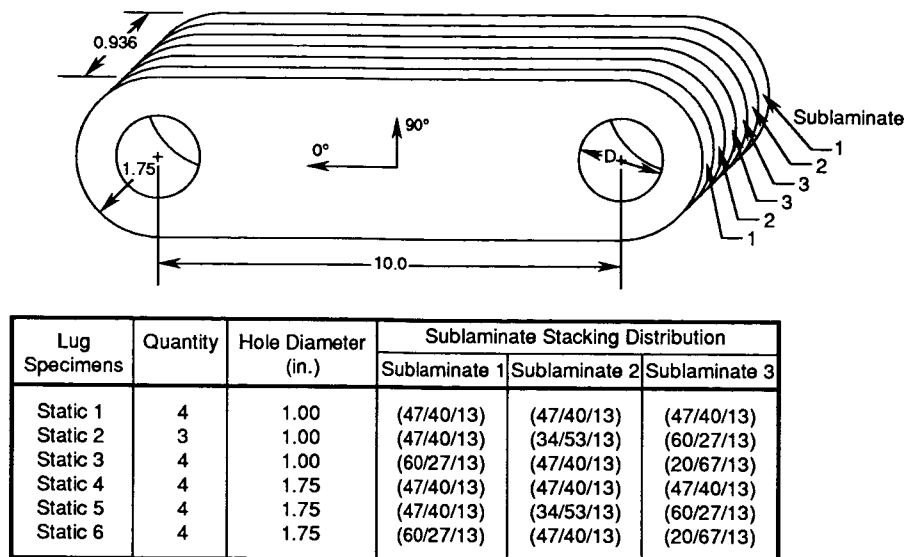
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Figure 2. Typical 2-D Lug and Pin Models



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Figure 3. Ply Volumes and Interface Areas With an Element



\* Sublamine Stacking Sequences are as follows:

(47/40/13): [0<sub>2</sub>/45<sub>2</sub>/90<sub>2</sub>/-45<sub>2</sub>/0<sub>2</sub>/45/0/-45/0<sub>2</sub>]s

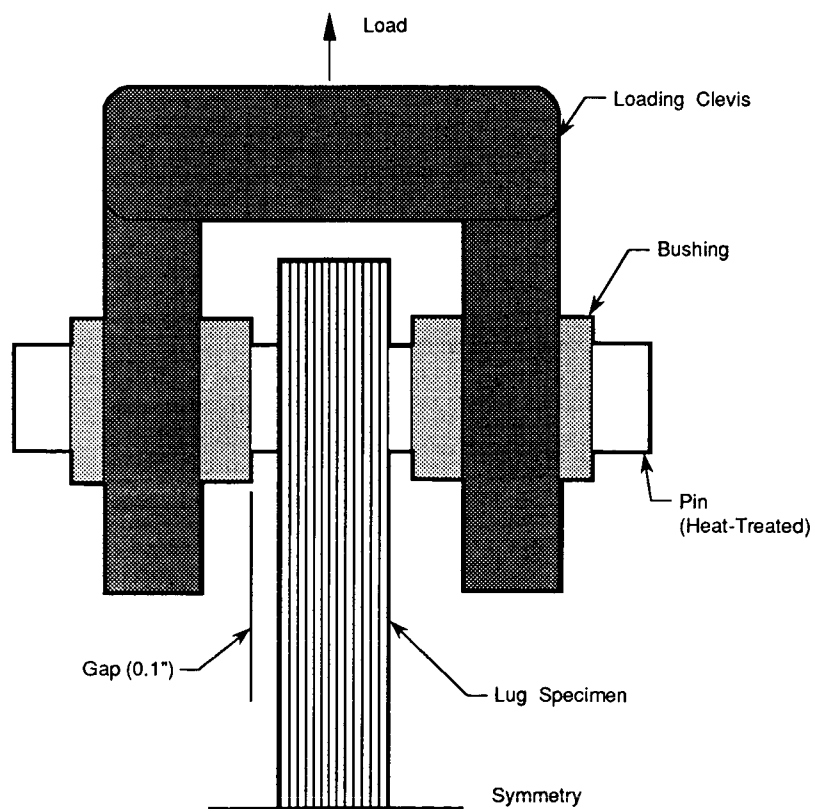
(34/53/13): [45<sub>2</sub>/0<sub>2</sub>/-45<sub>2</sub>/90<sub>2</sub>/45<sub>2</sub>/0/-45<sub>2</sub>/0]s

(60/27/13): [0<sub>2</sub>/45/0<sub>2</sub>/-45/0<sub>2</sub>/45/0<sub>2</sub>/-45/90<sub>2</sub>/0]s

(20/67/13): [45<sub>2</sub>/0/-45<sub>2</sub>/90/45<sub>2</sub>/0/-45<sub>2</sub>/90/45/0/-45]s

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Figure 4. Configuration and Stacking Sequence of Test Specimens



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Figure 5. Test Apparatus

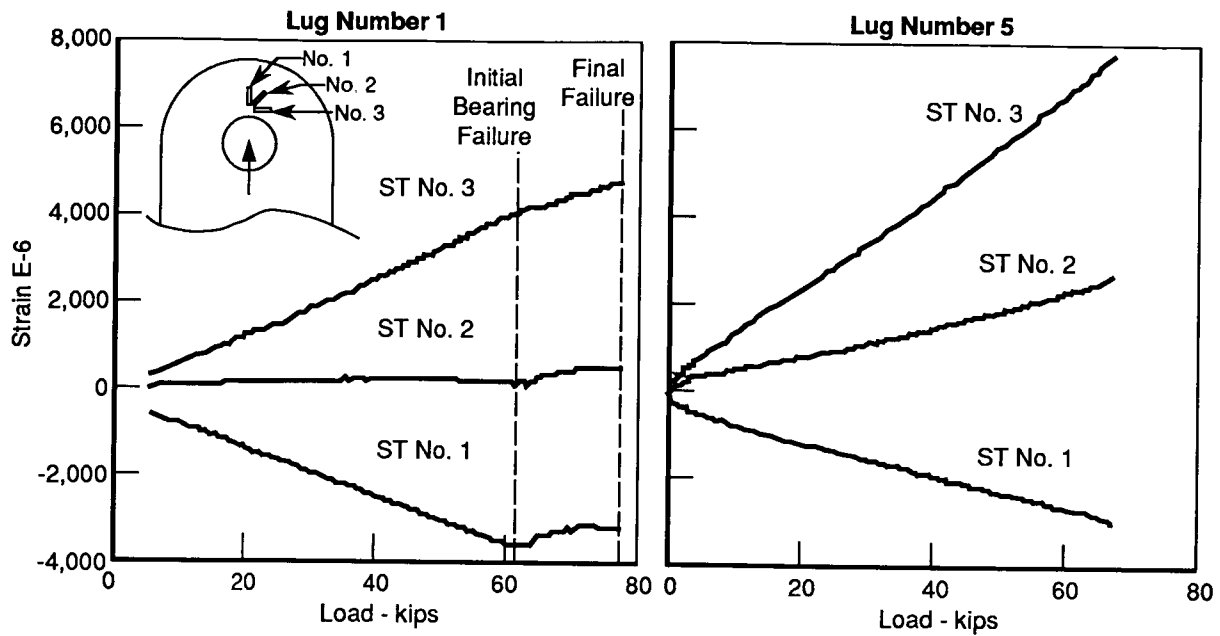


Figure 6. Load-Strain Curves for Composite Lugs

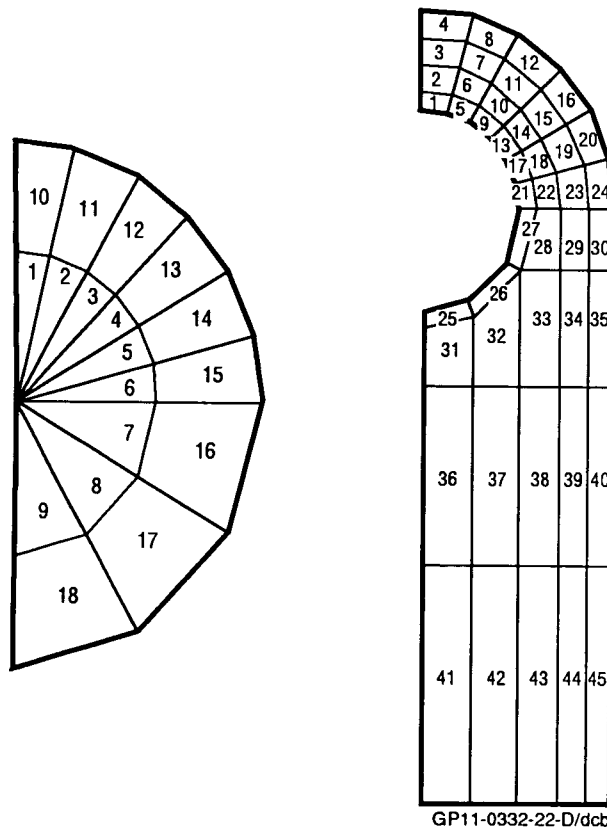
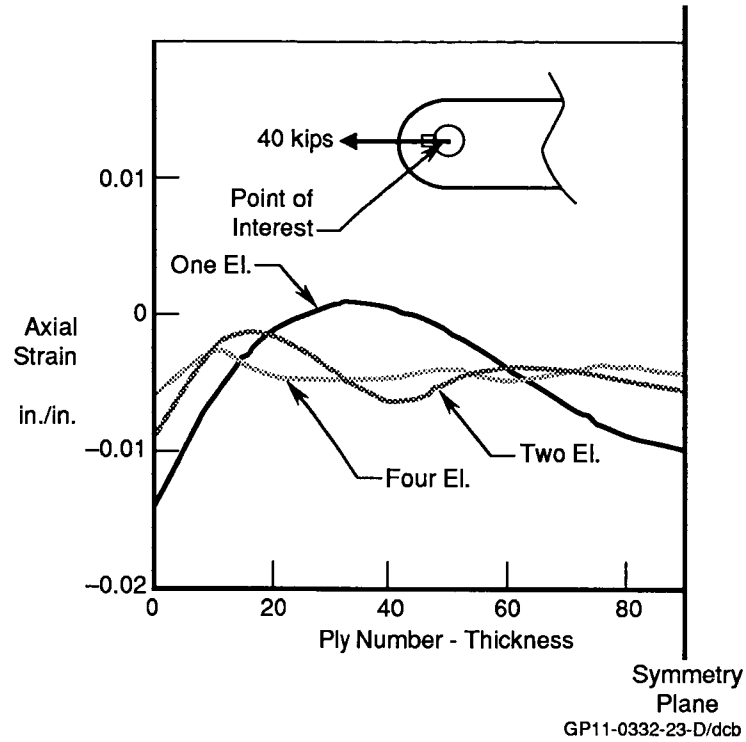


Figure 7. 2-D Metal for Lug and Pin Models



**Figure 8. Convergence Study for Lug Analysis**  
(For Lug Type 1)

| Lug Specimen | Quantity Each | Hole Dia. (in.) | Pred. Failure Load (kips) | Average Test Result (kips) | Standard Deviation |
|--------------|---------------|-----------------|---------------------------|----------------------------|--------------------|
| No. 1        | 4             | 1.00            | 64.2                      | 60.1 (76.7)*               | 4.3 (2.2)          |
| No. 2        | 3             | 1.00            | 65.9                      | 62.3 (74.3)*               | 6.9 (3.0)          |
| No. 3        | 4             | 1.00            | 57.3                      | 62.2 (74.7)*               | 7.0 (2.0)          |
| No. 4        | 4             | 1.75            | 66.9                      | 69.3 **                    | 3.1                |
| No. 5        | 4             | 1.75            | 66.5                      | 68.7 **                    | 2.9                |
| No. 6        | 4             | 1.75            | 61.5                      | 66.8**                     | 1.8.               |

\* Initial Bearing Failure Load (Final Failure Load)

\*\* Failure

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**Figure 9. Comparison of Experimental and Analytical Results**



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AN IMPROVED PLATE THEORY OF ORDER {1,2}  
FOR THICK COMPOSITE LAMINATES

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SUMMARY

A new {1, 2}-order theory is proposed for the linear elasto-static analysis of laminated composite plates. The basic assumptions are those concerning the distribution through the laminate thickness of the displacements, transverse shear strains and the transverse normal stress, with these quantities regarded as some weighted averages of their exact elasticity theory representations. The displacement expansions are linear for the inplane components and quadratic for the transverse component, whereas the transverse shear strains and transverse normal stress are respectively quadratic and cubic through the thickness. The main distinguishing feature of the theory is that all strain and stress components are expressed in terms of the assumed displacements prior to the application of a variational principle. This is accomplished by an a priori least-square compatibility requirement for the transverse strains and by requiring exact stress boundary conditions at the top and bottom plate surfaces. Equations of equilibrium and associated Poisson boundary conditions are derived from the virtual work principle. It is shown that the theory is particularly suited for finite element discretization as it requires simple  $C^0$ - and  $C^1$ -continuous displacement interpolation fields. Analytic solutions for the problem of cylindrical bending are derived and compared with the exact elasticity solutions and those of our earlier {1, 2}-order theory based on the assumed displacements and transverse strains.

INTRODUCTION

Designing of aerospace and ground-vehicle structures with thick-section organic-matrix composites is necessarily associated with the analytical modeling of the structural response and the prediction of failure under service loads. For such applications, viable analytical models are those that can properly account for transverse shear and transverse normal deformations. These effects, which are negligibly small in thin laminates, can be significant when the laminate thickness and the wavelength of loading are the same order of magnitude. Moreover, the reduced stiffness and strength in the transverse shear and transverse normal material directions can contribute to significant deformations and matrix dominated failure modes such as delamination and transverse cracking.

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The majority of finite element methods for composite laminates are based upon the  $\{1,0\}$ -order theories (commonly referred to as 1st-order theories). These account in some average sense for transverse shear but ignore the effects due to transverse normal deformations. The use of three-dimensional elements, including all three-dimensional effects, are generally computationally prohibitive even on a supercomputer, unless employed in some global-local fashion in a relatively small domain of interest. Although post-processing methods have been devised leading in some cases to adequate three-dimensional stress and strain recovery from the  $\{1,0\}$ -order theory computations, the static and dynamic response predictions obtained with this order of approximation are often inadequate for thick laminates (e.g., refs. (1-7) and references thereof).

Numerous efforts to generate laminate theories of higher-order that also include transverse normal deformations failed to produce sufficiently accurate and computationally suitable formulations to model thick-section composites. Recently, Tessler (ref. 1) proposed a  $\{1,2\}$ -order theory for the elasto-static analysis of homogeneous orthotropic plates which was demonstrated to be ideally suited for finite element approximations. Subsequent developments included the analytic and finite element analyses of elastic beams (refs. 2-4), an extension to the general  $\{1,2\}$ -order orthotropic shell theory (ref. 5), and an extension to laminated composite plates for elasto-statics (ref. 6) and elasto-dynamics (ref 7). These theories assume linear inplane displacements and a quadratic transverse displacement across the thickness. The key aspect in the approximation is the independent expansions for transverse strains which allow exact stress boundary conditions at the top and bottom plate faces to be satisfied. These strains are also least-square compatible with those derived directly from the displacement assumptions.

In this paper we propose a new  $\{1,2\}$ -order theory for the linear elasto-static analysis of laminated composite plates following the methodology established in (ref. 6). The present formulation departs from (ref. 6) in that the transverse normal stress is independently expanded across the laminate thickness instead of the transverse normal strain. The transverse shear strains are taken to be parabolic satisfying traction-free boundary conditions. The transverse normal stress is assumed to be a cubic function through the thickness, also satisfying the consequence of zero shear tractions at the top and bottom laminate surfaces; specifically, the vanishing on those surfaces of the transverse normal stress gradient taken with respect to the thickness coordinate. These latter conditions are exact according to three-dimensional elasticity theory when the body and inertial forces are neglected. The resulting equilibrium equations are associated with exclusively Poisson boundary conditions. As its predecessor theory, the present theory requires only  $C^0$ - and  $C^1$ -continuous displacement interpolation fields and thus lends itself well to the development of simple, robust, and computationally efficient plate formulations similar to Mindlin-type elements (refs. (9-11)).

In assessing the accuracy of the proposed theory, we resort to the problem of cylindrical bending of an infinite laminated plate for which an exact elasticity solution is available (ref. 8). We compare the results of the present theory for several lamination patterns and span-to-thickness ratios with those of our previous  $\{1,2\}$ -order laminate plate theory (ref. 6) and the exact solutions.

## NOMENCLATURE

|                              |                                                                           |
|------------------------------|---------------------------------------------------------------------------|
| $A_{ij}$                     | membrane plate rigidities                                                 |
| $C_{ij}^{(k)}$               | elastic stiffness coefficients for the $k^{\text{th}}$ ply                |
| $B_{ij}$                     | membrane-bending coupling plate rigidities                                |
| $D_{ij}$                     | bending plate rigidities                                                  |
| $G_{ij}$                     | transverse shear plate rigidities                                         |
| $C^0$                        | continuous functions with discontinuous derivatives at element interfaces |
| $C^{-1}$                     | continuous functions that are discontinuous at element interfaces         |
| $2h$                         | plate thickness                                                           |
| $N_{ij}, Q_i$                | inplane and transverse shear force resultants                             |
| $M_{ij}$                     | bending moment resultants                                                 |
| $q^+$                        | applied transverse load at $z=h$                                          |
| $u, v$                       | midplane displacements along x and y axes                                 |
| $u_x, u_y, u_z$              | Cartesian displacement components                                         |
| $w, w_i$                     | components of the transverse displacement                                 |
| $x, y$ and $z$               | inplane and transverse coordinates                                        |
| $\delta$                     | variational operator                                                      |
| $\epsilon_{ij}, \kappa_{ij}$ | strain and curvature components                                           |
| $\theta_i$ ( $i=x, y$ )      | bending rotations about x and y axes                                      |
| $\xi$                        | dimensionless thickness coordinate                                        |
| $\sigma_{ij}, \tau_{ij}$     | stress components                                                         |

## {1, 2}-KINEMATIC PLATE ASSUMPTIONS

In a laminated composite plate constructed of perfectly bonded orthotropic plies whose transverse constitutive properties do not differ appreciably, the displacement vector  $\mathbf{u} = (u_x, u_y, u_z)$  can be approximated with functions that vary continuously across the total thickness. The lowest-order displacement expansions accounting for transverse shear and transverse normal deformations involve linear thickness variations for the inplane components  $u_x$  and  $u_y$  and a quadratic one for the transverse displacement  $u_z$  (ref. 6) (i.e., {1,2}-order theory):

$$\begin{aligned}
u_x(x, y, z) &= u(x, y) + h\xi \theta_y(x, y), & u_y(x, y, z) &= v(x, y) + h\xi \theta_x(x, y) \\
u_z(x, y, z) &= w(x, y) + \xi w_1(x, y) + (\xi^2 - 1/5)w_2(x, y)
\end{aligned} \tag{1}$$

where  $\xi=z/h \in [-1, 1]$  is the dimensionless thickness coordinate and  $\xi=0$  identifies the reference midplane position;  $u(x,y)$  and  $v(x,y)$  are the midplane displacements along the  $x$  and  $y$  axes, and  $\theta_x(x,y)$  and  $\theta_y(x,y)$  are the rotations of the normal about the  $x$  and  $y$  axes (see fig. 1). The  $w_1$  and  $w_2$  variables can be interpreted as the normalized strain and curvature in the thickness direction:

$$w_1/h = u_{z,z}|_{z=0}, \quad w_2/h^2 = \frac{1}{2}u_{z,zz} \tag{1.1}$$

The 3-D Hooke's law for any  $k$ th orthotropic ply is expressed in the *mixed* form<sup>2</sup>

$$\begin{Bmatrix} \sigma_{xx} \\ \sigma_{yy} \\ \tau_{xy} \\ \epsilon_{zz} \\ \tau_{yz} \\ \tau_{xz} \end{Bmatrix}^{(k)} = \begin{bmatrix} \hat{C}_{11} & \hat{C}_{12} & \hat{C}_{16} & R_{13} & 0 & 0 \\ \hat{C}_{12} & \hat{C}_{22} & \hat{C}_{26} & R_{23} & 0 & 0 \\ \hat{C}_{16} & \hat{C}_{26} & \hat{C}_{66} & R_{63} & 0 & 0 \\ -R_{13} & -R_{23} & -R_{63} & S_{33} & 0 & 0 \\ 0 & 0 & 0 & 0 & C_{44} & C_{45} \\ 0 & 0 & 0 & 0 & C_{45} & C_{55} \end{bmatrix}^{(k)} \begin{Bmatrix} \epsilon_{xx} \\ \epsilon_{yy} \\ \gamma_{xy} \\ \sigma_{zz} \\ \gamma_{yz} \\ \gamma_{xz} \end{Bmatrix} \tag{2}$$

with

$$\begin{aligned}
\hat{C}_{ij}^{(k)} &= C_{ij}^{(k)} - C_{i3}^{(k)} C_{j3}^{(k)} / C_{33}^{(k)} \\
R_{i3}^{(k)} &= C_{i3}^{(k)} / C_{33}^{(k)}, \quad S_{33}^{(k)} = 1/C_{33}^{(k)} \quad (i = 1, 2, 6)
\end{aligned} \tag{2.1}$$

where  $C_{ij}^{(k)}$  are the elastic stiffness coefficients corresponding to the  $x$ - $y$  coordinates (ref. 12).

The components of strain and stress can be expressed in terms of the plate strain and curvature variables that are independent of the thickness coordinate:

---

<sup>2</sup> The variables superscribed with the  $k$  index are ply-dependent, whereas those without the  $k$  index are some average representations across the laminate thickness and thus are independent of the individual ply properties.

$$\begin{aligned}
\epsilon_o^T &= [\epsilon_{xo}, \epsilon_{yo}, \epsilon_{zo}, \gamma_{xyo}] = [u_x, v_y, w_1/h, u_y + v_x] \\
\kappa_o^T &= [\kappa_{xo}, \kappa_{yo}, \kappa_{zo}, \kappa_{xyo}] = [\theta_{yx}, \theta_{xy}, w_2/h^2, \theta_{xx} + \theta_{yy}] \\
\gamma_o^T &= [\gamma_{xzo}, \gamma_{yzo}] = [w_x + \theta_y, w_y + \theta_x]
\end{aligned} \tag{3}$$

The inplane strains, obtained from eq. (1) in accordance with elasticity theory, are

$$\epsilon_{xx} = \epsilon_{xo} + z \kappa_{xo}, \quad \epsilon_{yy} = \epsilon_{yo} + z \kappa_{yo}, \quad \gamma_{xy} = \gamma_{xyo} + z \kappa_{xyo} \tag{4}$$

The present theory departs from that in ref. 6 only in the manner in which the transverse normal strain and stress are developed. Here, we independently expand the transverse shear strains and the transverse normal stress. The motivation for the latter assumption stems from a careful examination of a series of 3-D exact elasticity stress solutions (ref. 8) for a cylindrical bending problem (see fig. 2). Figure 3 shows 3-D solutions for the transverse shear and transverse normal stresses corresponding to the four distinct laminations: [0], [30/-30]s, [0/90]s and [0/90]. It is quite revealing that the form of the transverse normal stress remains relatively consistent from lamination to lamination whereas the transverse shear stress changes dramatically. These elasticity results clearly suggest that an independent, continuously varying expansion for the transverse normal stress should be an improvement over an analogous expansion for the transverse normal strain as used in (ref. 6). On the other hand, it is unlikely that one would gain any advantage by expanding the transverse shear stresses in a continuous manner across the laminate over that concerning the transverse shear strain expansions (ref. 6) since the solutions for these stresses (fig. 3) cannot be reproduced with a relatively low order approximation such as a parabola.

The transverse normal stress and transverse shear strains are expanded independently across the total laminate thickness as<sup>3</sup>

$$\sigma_{zz} = \sum_{n=0}^3 \sigma_{zn} \xi^n, \quad \gamma_{iz} = \sum_{n=0}^2 \gamma_{in} \xi^n \quad (i = x, y) \tag{5}$$

The expansion coefficients  $\sigma_{zn} = \sigma_{zn}(x, y)$  and  $\gamma_{in} = \gamma_{in}(x, y)$  are determined by requiring the stress field to satisfy exact traction conditions at the top and bottom plate surfaces

$$\tau_{iz}^{(k)}(x, y, \pm h) = \sigma_{zz}^{(k)}(x, y, \pm h) = 0 \quad (i = x, y) \tag{6}$$

along with the transverse strains to be least-square compatible with those obtained directly from elasticity theory using our approximations for the displacements (1):

---

<sup>3</sup> The present approach and that in ref. 6 are equivalent for homogeneous plates (ref. 1).

$$\begin{aligned} \text{minimize} \quad & \int_{-h}^h [\epsilon_{zz}^{(k)} - u_{z,z}]^2 dz \\ \text{minimize} \quad & \int_{-h}^h [\gamma_{iz} - (u_{z,i} + u_{i,z})]^2 dz \quad (i = x, y) \end{aligned} \quad (7)$$

where the minimizations are performed with respect to the  $\sigma_{zn}$  and  $\gamma_{in}$  variables. The resulting transverse normal stress and transverse shear strains can be expressed as

$$\sigma_{zz} = \sigma_{zo}(\epsilon_o, \kappa_o) + \sigma_{zi}(\epsilon_o, \kappa_o)(\xi - \xi^3/3), \quad \gamma_{iz} = \frac{5}{4}(1 - \xi^2)\gamma_{io} \quad (i = x, y) \quad (8)$$

Application of the virtual work principle results in the 10<sup>th</sup>-order plate equations of equilibrium and associated Poisson boundary conditions. The principle also serves as a variational framework for finite element approximations. The resulting two-dimensional variational statement has the form

$$\int_{S_+} [N^T \delta \epsilon_o + M^T \delta \kappa_o + Q^T \delta \gamma_o] dxdy - \delta W_e(u, v, w, \theta_x, \theta_y, w_1, w_2) = 0 \quad (9)$$

where  $\delta W_e$  denotes the virtual work of external forces;  $N=\{N_{ij}\}$ ,  $M=\{M_{ij}\}$  and  $Q=\{Q_i\}$  are vectors of the plate stress resultants which are related to the plate strains (eqs. (3)) via the constitutive relations

$$\begin{Bmatrix} N \\ M \\ Q \end{Bmatrix} = \begin{bmatrix} A & B & 0 \\ B^T & D & 0 \\ 0 & 0 & G \end{bmatrix} \begin{Bmatrix} \epsilon_o \\ \kappa_o \\ \gamma_o \end{Bmatrix} \quad (10)$$

where  $A=[A_{ij}]$ ,  $B=[B_{ij}]$ ,  $D=[D_{ij}]$  and  $G=[G_{ij}]$  are the plate constitutive matrices.

#### REMARKS ON FEM APPROXIMATIONS

The variational principle (9) provides a convenient framework for developing efficient plate bending elements for the analysis of thin and thick composite laminates. In eq. (9), the kinematic variables  $u$ ,  $v$ ,  $w$ ,  $\theta_x$  and  $\theta_y$  possesses spatial derivatives that do not exceed order one, thus requiring only  $C^0$ -continuous finite element trial functions;  $w_1$  and  $w_2$  do not have spatial gradients thus needing only  $C^1$  approximations.

The simplest laminate plate element, which is also the most desirable from the standpoint of adaptive mesh refinement, is a three-node anisoparametric triangle (ref. 6). The element employs linear (3-node) parametric functions for  $u$ ,  $v$ ,  $\theta_x$  and  $\theta_y$ , an anisoparametric quadratic field for  $w$  and uniform fields for  $w_1$  and  $w_2$ . The latter variables only contribute two degrees-of-freedom per element; in static problems, they can be conveniently condensed out at the element level using static condensation.

## RESULTS AND DISCUSSION

The present theory will be qualitatively assessed with respect to its application to the problem of cylindrical bending of an infinite carbon/epoxy laminate subjected to a sinusoidal transverse pressure  $q^+ = q_0 \sin(\pi x/L)$ . The exact elasticity solution for this problem was first determined by Pagano (ref. 8). We shall consider an orthotropic [0] laminate, two symmetric laminates — a cross-ply [0/90]<sub>s</sub> laminate and an angle-ply [30/-30]<sub>s</sub> laminate — and an antisymmetric cross-ply laminate, [0/90]. The ply material properties are taken as

$$\begin{aligned} E_L &= 25 \times 10^6 \text{ psi}, & E_T &= 10^6 \text{ psi}, & G_{LT} &= 0.5 \times 10^6 \text{ psi} \\ G_{TT} &= 0.2 \times 10^6 \text{ psi}, & \nu_{LT} &= \nu_{TT} = 0.25 \end{aligned} \quad (11)$$

where L and T denote the longitudinal and transverse ply material directions, respectively.

As noted previously, this theory differs from ref. 6 only in the manner in which the transverse normal stress and strain are approximated. Thus, it is only natural to expect that the differences in the predictions with the two theories will mostly affect these transverse normal variables. Indeed, the displacement predictions by both theories are virtually identical, and so are the inplane and transverse shear stresses and strains.

Figure 4 depicts the percent error in the maximum deflection computed at the midplane of the laminate versus the length-to-thickness ratio  $L/2h$ . The results, which pertain to the present and ref. 6 theories, clearly show that as far as the deflection predictions are concerned, the engineering accuracy (i.e., error  $\leq 5\%$ ) is attained for laminates with the ratio  $L/2h \geq 4$ . These results demonstrate the adequacy of the overall plate stiffness representation.

Figures 5 through 8 compare the stress and strain thickness distributions for thin ( $L/2h=40$ ) and thick ( $L/2h=4$ ) laminates obtained with the present theory (designated as HOT-S), our previous {1,2}-theory (HOT-E, ref. 6), and the exact solutions obtained in this effort using the 3-D elasticity approach (ref. 8). Figure 5 shows the transverse normal stress distributions across the laminate thickness. Note that while both HOT-E and HOT-S are nearly equally accurate in the [30/-30]<sub>s</sub> lamination predictions — with HOT-E exhibiting a slight discontinuity at the interface between 30 deg. and -30 deg. plies — the HOT-S prediction for the cross-ply [0/90] laminate is much superior. The transverse normal strain distributions are depicted in fig. 6, where the present theory exhibits proper discontinuous character at the ply interfaces. Except on the bottom surface, both HOT-E and HOT-S produce appreciable errors in the [0/90] thick laminate.

The transverse shear stress ( $\tau_{xz}$ ) distributions are compared in fig. 7. The HOT-S and HOT-E stresses are computed by integrating appropriate inplane stress gradients in the 3-D equations of equilibrium (ref. 6). Throughout, these predictions compare well with the exact elasticity solutions. Figure 8 depicts thickness distributions of the normal stress ( $\sigma_{xx}$ ). These show that for the [30/-30]<sub>s</sub> thick laminate the stresses in the outer plies are underestimated.



This is naturally the result of linear assumptions for the inplane displacements (1). Interestingly, the  $\sigma_{xx}$  results for the [0/90] thick laminate are excellent.

### CONCLUDING SUMMARY

In this paper we have discussed a new {1, 2}-order theory for the elasto-static analysis of laminated composite plates in which the assumed variables are the components of the displacement vector, the transverse shear strains and the transverse normal stress. The transverse stresses satisfy exact stress boundary conditions at the top and bottom plate surfaces. The virtual work principle produces a set of equilibrium equations and associated Poisson boundary conditions. As was the previous theory, this theory is particularly suited for finite element approximation with simple  $C^0$ - and  $C^1$ -continuous displacement interpolation fields. The analytic solutions - obtained for a wide range of laminations and thicknesses - showed that the theory is applicable to thin and thick laminated composites and has some advantages over our previous {1, 2}-order theory.

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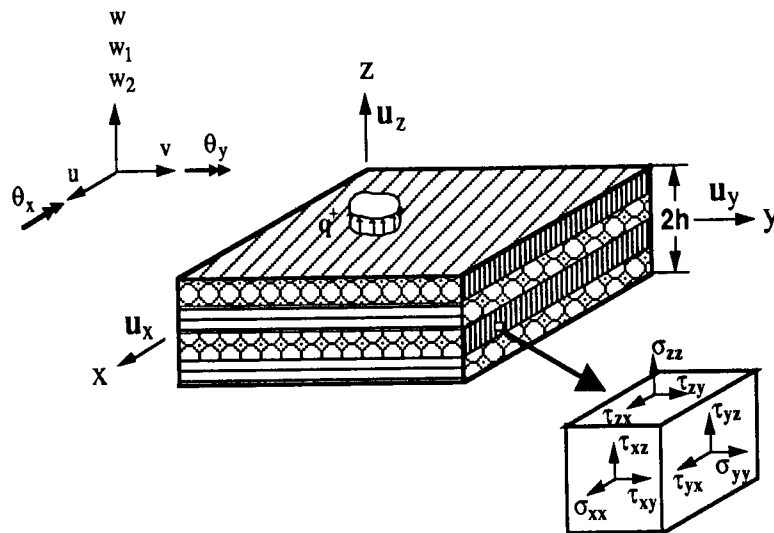


Figure 1. Notation for {1,2}-order plate theory.

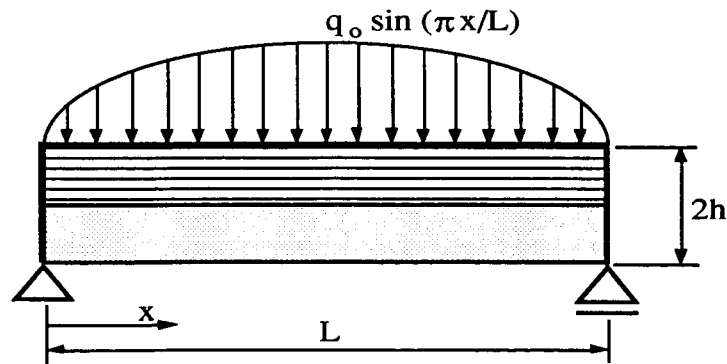


Figure 2. Cylindrical bending of infinite laminated plate.

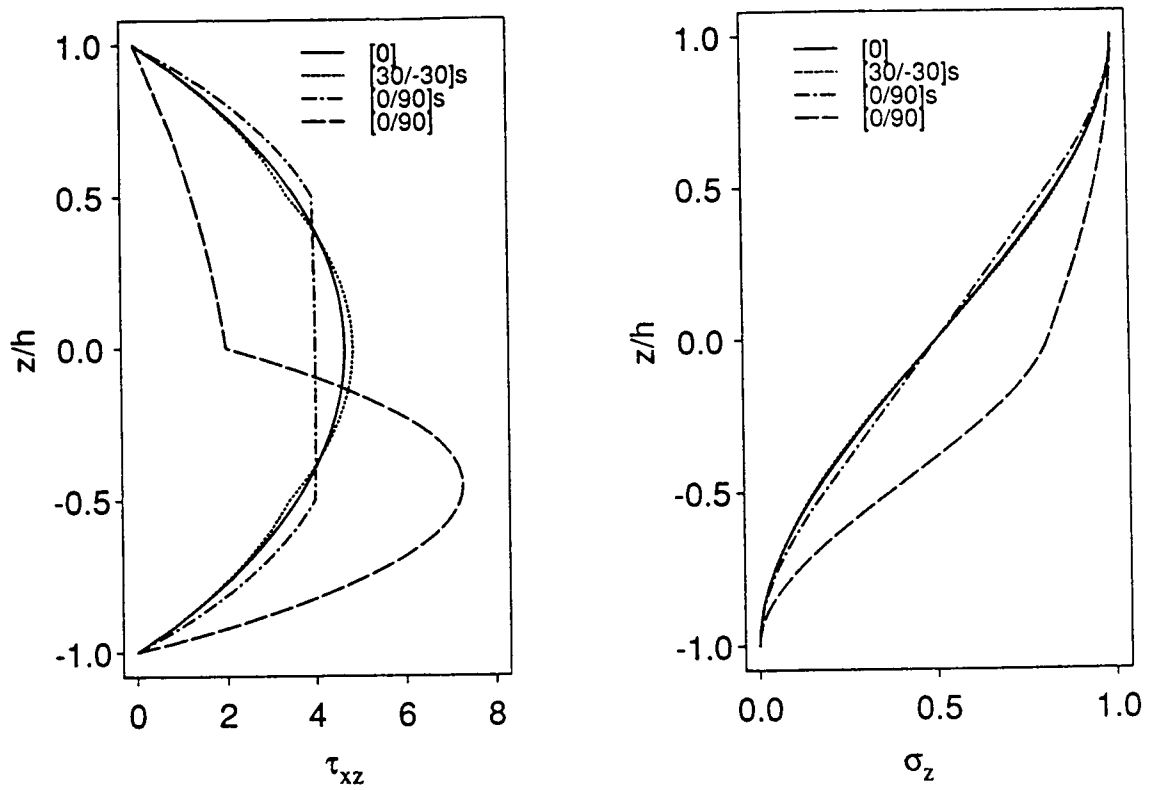


Figure 3. Exact distributions of  $\tau_{xz}(0, z)$  and  $\sigma_z(L/2, z)$  across thickness in Gr/Ep laminates,  $L/2h=10$ .

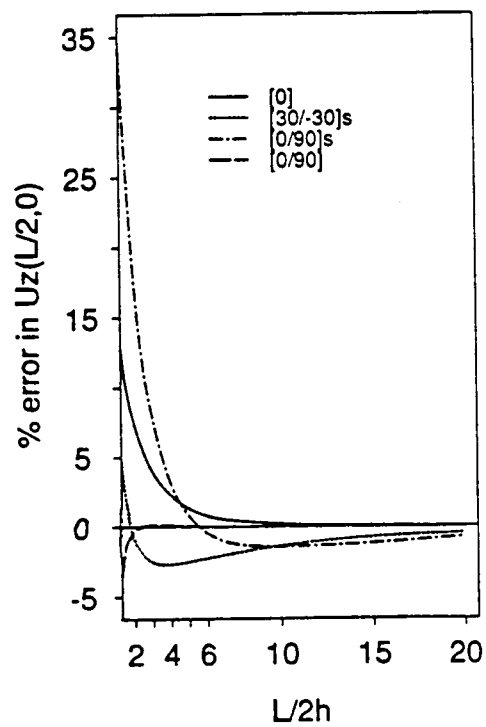


Figure 4. Percent error in maximum midplane deflection vs.  $L/2h$  ratio for various Gr/Ep laminates.

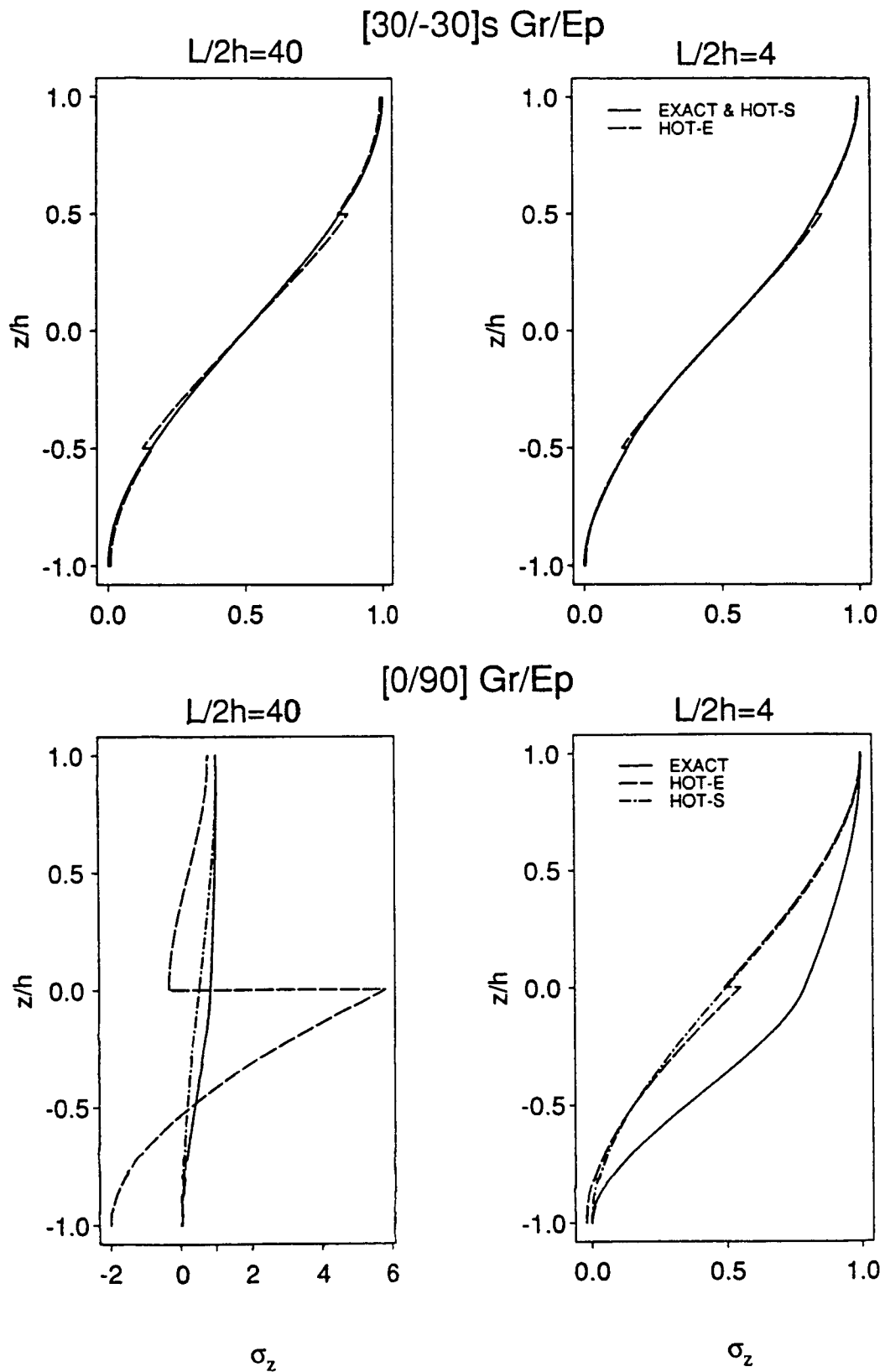


Figure 5. Distributions of  $\sigma_z(L/2, z)$  across thickness in Gr/Ep laminates;  $L/2h = 40, 4$ .

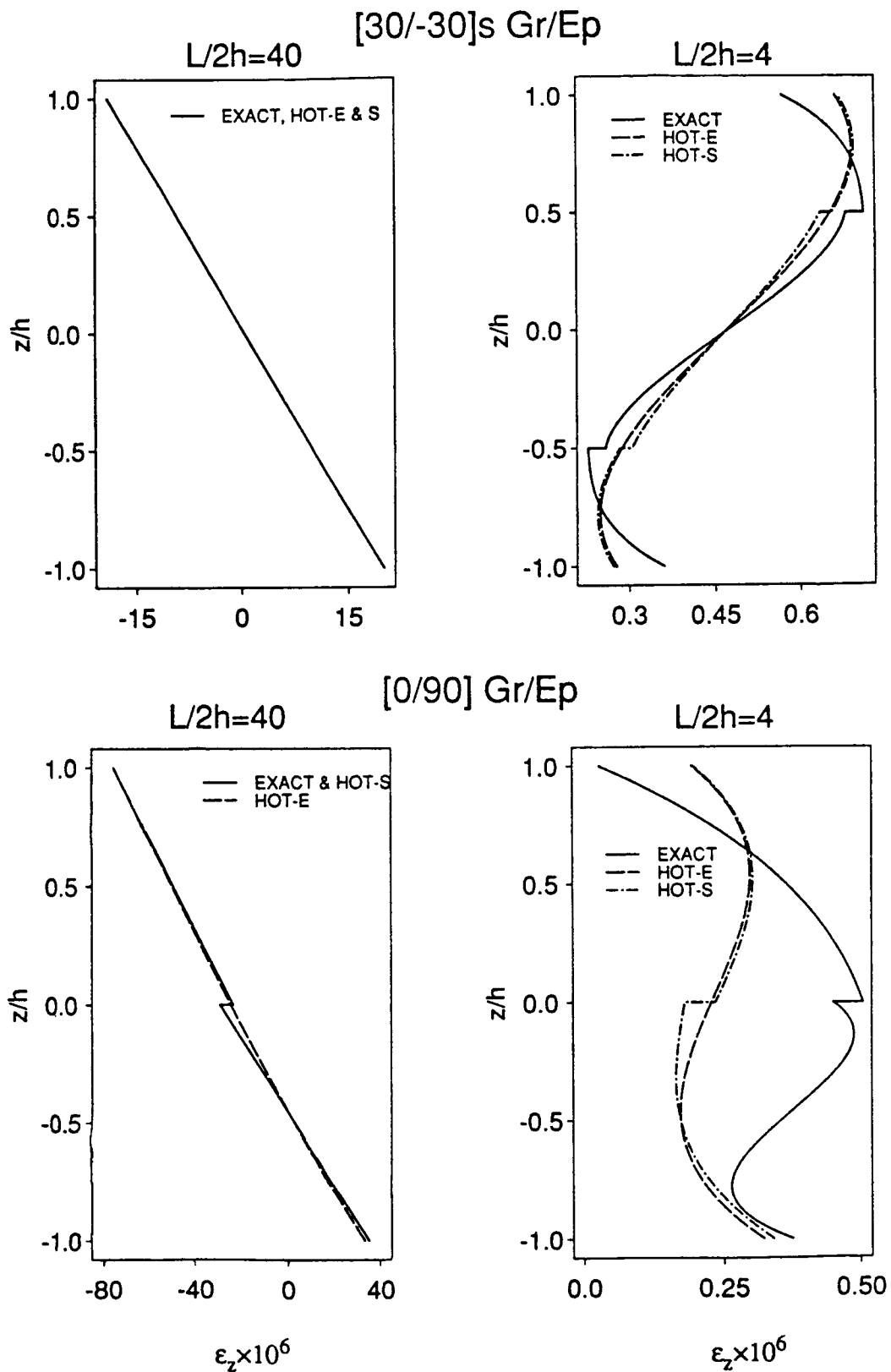


Figure 6. Distributions of  $\epsilon_z(L/2, z)$  across thickness in Gr/Ep laminates;  $L/2h = 40, 4$ .

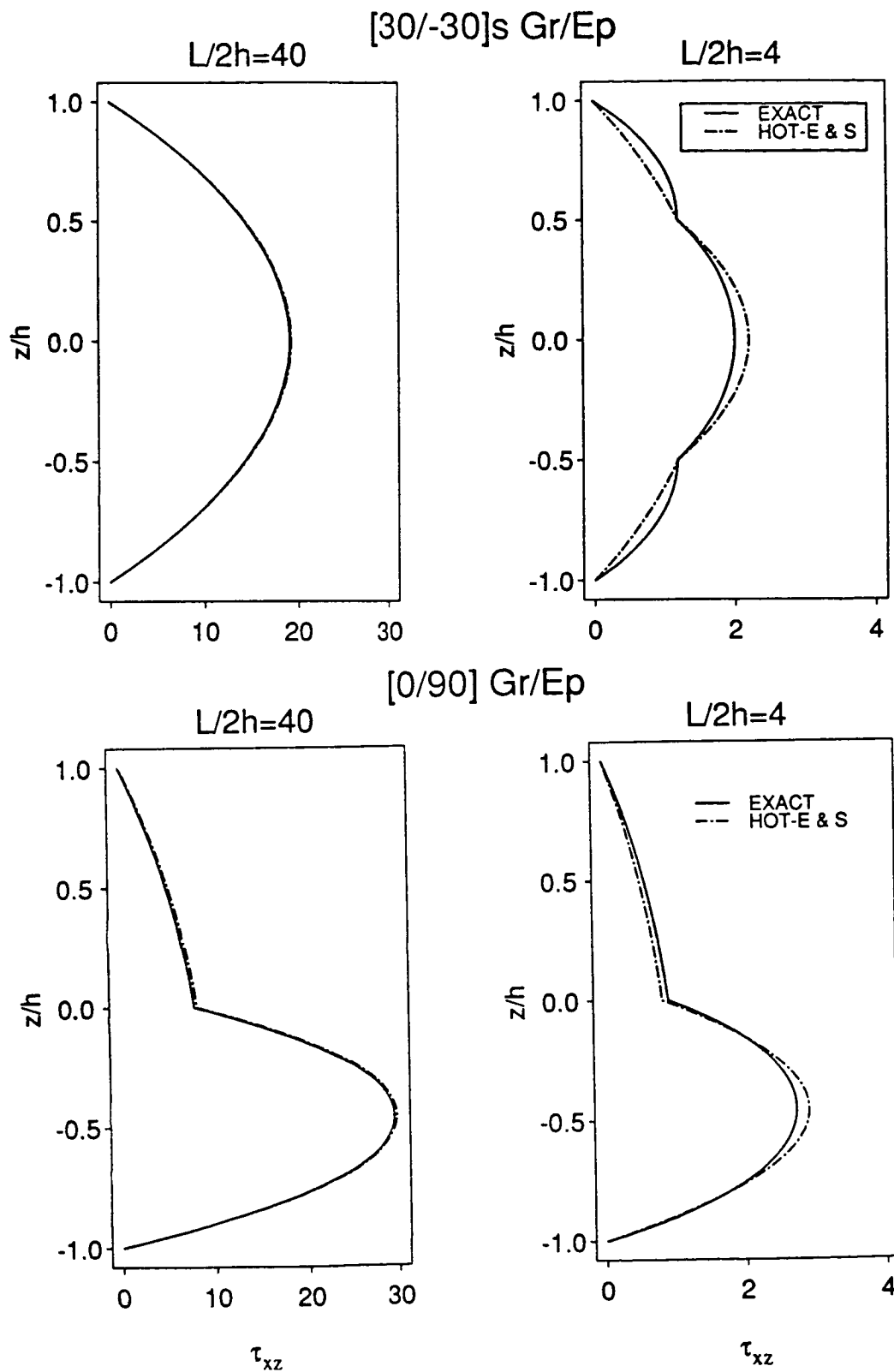


Figure 7. Distributions of  $\tau_{xz}(0, z)$  across thickness in Gr/Ep laminates;  $L/2h = 40, 4$ .

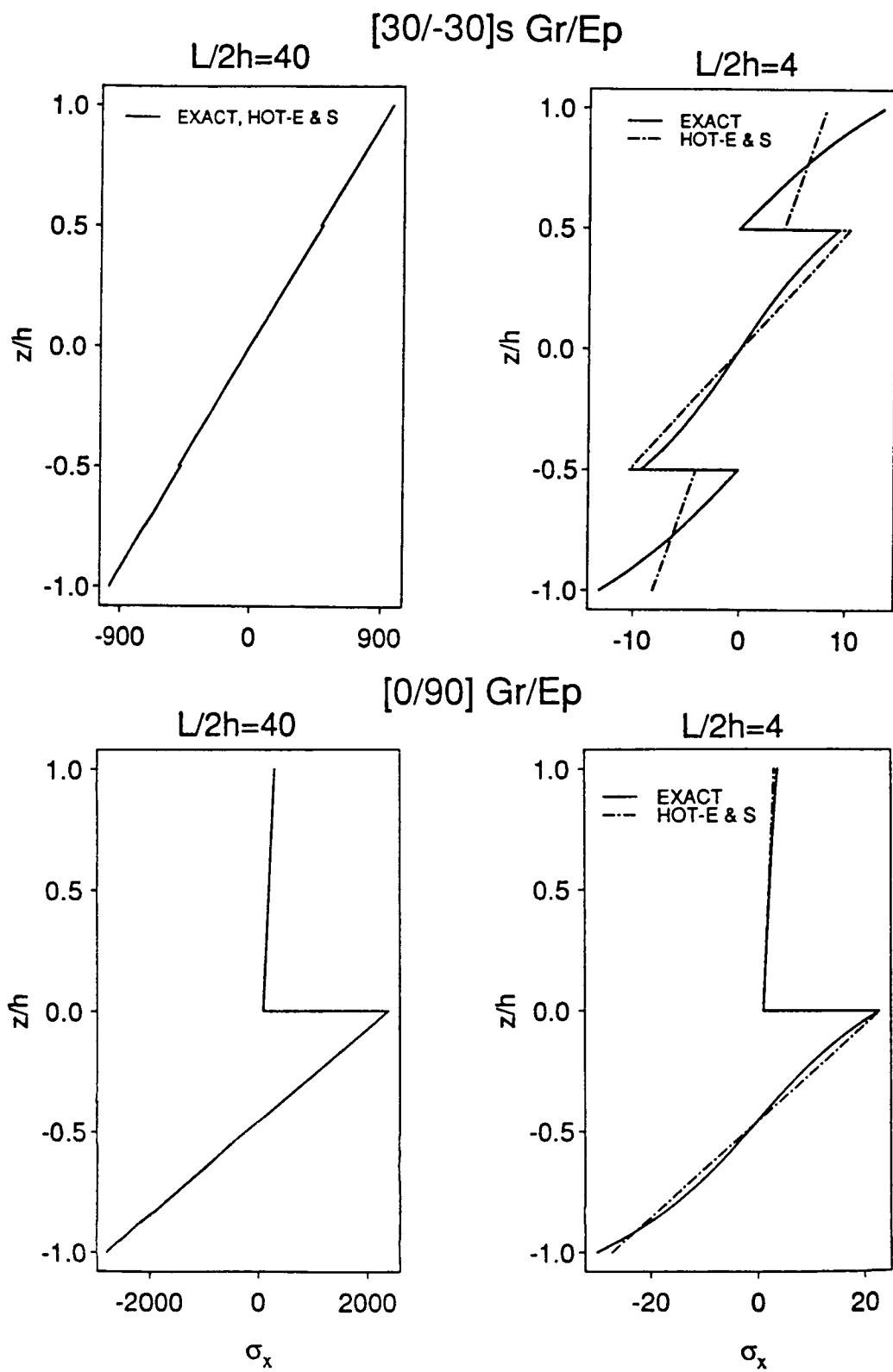


Figure 8. Distributions of  $\sigma_x(L/2, z)$  across thickness in Gr/Ep laminates;  $L/2h = 40, 4$ .

FAILURE ANALYSIS OF THICK COMPOSITE CYLINDERS  
UNDER EXTERNAL PRESSURE<sup>1</sup>

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## SUMMARY

Failure of thick section composites due to local compression strength and overall structural instability is treated. Effects of material nonlinearity, imperfect fiber architecture, and structural imperfections upon anticipated failure stresses are determined. Comparisons with experimental data for a series of test cylinders are described.

Predicting the failure strength of composite structures requires consideration of stability and material strength modes of failure using linear and nonlinear analysis techniques. Material strength prediction requires the accurate definition of the local multiaxial stress state in the material. An elasticity solution for the linear static analysis of thick anisotropic cylinders and rings is used herein to predict the axisymmetric stress state in the cylinders. Asymmetric nonlinear behavior due to initial cylinder out of roundness and the effects of end closure structure are treated using finite element methods.

It is assumed that local fiber or ply waviness is an important factor in the initiation of material failure. An analytical model for the prediction of compression failure of fiber composites, which includes the effects of fiber misalignments, matrix inelasticity and multiaxial applied stresses is used for material strength calculations. Analytical results are compared to experimental data for a series of glass and carbon fiber reinforced epoxy cylinders subjected to external pressure. Recommendations for pretest characterization and other experimental issues are presented. Implications for material and structural design are discussed.

## INTRODUCTION

The design, analysis and failure prediction of thick fiber composite cylinders is made complex at the materials level by the multiplicity of material characteristics influencing the failure mechanism, and at the structural level by the importance of three

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<sup>1</sup> A portion of this work was performed under Subcontract Number 11K-57957C for Martin Marietta Energy Systems at Oak Ridge National Laboratory



dimensional stress states in anisotropic materials. Basic constituent properties such as relative stiffnesses and strength of the fiber and matrix directly affect the mode and stress at material failure. Fabrication related material variables, such as fiber misalignments, ply curvatures, disbonding and resin rich (starved) areas, are also important factors in the compressive failure mechanics of the material. A compression strength model which addresses many of these key material parameters is discussed below. Unlike a semi-empirical phenomenological polynomial failure criterion, this model is physically based on an understanding of the local failure mechanisms. An approach for predicting structural failure of unstiffened hydrostatically loaded composite cylinders, which considers material failure, elastic stability, and nonlinear collapse modes of failure, was developed to evaluate alternative designs. The failure prediction methodology was applied to a series of cylinders fabricated and tested during the DARPA Composite Materials for Future Submarines (CMFS) project. Analysis and experimental failure results for two of the cylinders are discussed below.

### COMPRESSIVE STRENGTH OF FIBER COMPOSITES

It has been widely accepted that the axial compressive failure mode of a unidirectional composite is local instability. Microbuckling was first proposed as a failure mechanism by Dow and Gruntfest [1]. This work suggested that small wavelength fiber buckling occurs in a fashion analogous to buckling of a column supported by an elastic foundation. Rosen [2] provided the initial analytical solution for the instability load by treating the reinforcement and matrix as a layered two dimensional medium. Two possible failure modes were considered: one in which adjacent fibers buckle in opposite directions (extensional mode); and one in which adjacent fibers buckle in phase with one another. For composites used for structural applications, the shear mode occurs at a lower applied load and is the governing failure mechanism. For elastic material behavior, the shear mode microbuckling result predicts instability failure at the same stress as the shear kinking mode examined subsequently by Budiansky [3]. Compressive failure for the shear microbuckling and shear kinking modes occurs at a stress numerically equal to the axial shear modulus of the unidirectional composite. This level of stress is not usually achieved in experiments, although the observed failure mode generally agrees with the analytical prediction.

Various investigators have included fiber out-of-straightness or misalignment, inelastic matrix response, weak interfaces and low strain to failure of the fiber to explain the discrepancy between experimental and basic theoretical results. In the model used to predict material failure in the thick walled cylinders, the observed nonlinear shear stress-strain behavior typical of glass and carbon fiber reinforced epoxies has been attributed to nonlinear matrix behavior. If significant shearing stresses are present in the matrix, due to applied loads or generated by local perturbations in fiber straightness, the matrix shear stiffness will be less than the initial

elastic value. For a general three dimensional state of stress, the reduced matrix tangent shear modulus can be computed based upon the second invariant of the deviatoric stress tensor. The reduced matrix shear stiffness results in a reduced composite tangent shear modulus and, hence, lower compressive strengths.

Matrix nonlinearity, imperfect interfaces, and fiber misalignment, while important effects, may not completely account for the discrepancies found between experimental strengths and strengths predicted based upon local instability theories. The failure strength, and failure mode of a fiber composite material is a function of each constituent's stiffness and failure strength. While the primary mode of failure for composites loaded in axial compression appears to be a local instability, it is important to recognize that accurate predictions of composite compressive strength must consider all possible failure modes. A graphical representation of the heterogeneous failure surface of a unidirectional fiber composite can be constructed for up to three components of applied load. Such a failure surface is presented in figure 1. For a uniaxial load (i.e., see also figure 2), the initiating failure mechanism is calculated to be fiber failure. As the applied transverse normal and shear stresses are increased from zero, microbuckling becomes the predominate mode of failure. For very high values of applied transverse normal and shear stresses, matrix failure is predicted to initiate overall ply failure. Hence, determination of the compressive strength of a fiber composite requires definition of the local multi-dimensional stress or strain state in principal material coordinates.

#### CYLINDER FAILURE ANALYSIS APPROACH

This section outlines a methodology for the failure prediction of hydrostatically loaded composite cylinders. Two levels of variables likely to affect the collapse strength of unstiffened composite cylinders were investigated: local ply (fiber bundle) waviness and global cylinder out of roundness. Material strength prediction is based upon inelastic microbuckling theory. The structural behavior of the cylinders was predicted using a closed form elasticity solution to solve for axisymmetric response and the finite element method to solve for asymmetric response.

A computerized solution for the linear static and linear viscoelastic analysis of thick axisymmetric anisotropic cylinders and rings was used to determine cylinder deflections, nominal fiber bundle (ply) stresses, and strains. This structural model is based upon linear elasticity theory [4, 5] and assumes that the axial strain response of the cylinder is independent of axial location. Discontinuity stresses developed near the end closures are not included. Any linear combination of internal and external pressure, end load, torque, and uniform temperature change may be defined as load inputs. Time dependent linear viscoelastic matrix behavior is permitted by the model but will not be addressed here.

Unidirectional composite (ply) properties were computed using the properties of the fiber and matrix and the composite cylinders assembly model [6]. Composite material properties were predicted using the three dimensional mini-mechanics theory outlined in reference [7]. To determine a baseline failure pressure for each cylinder design, computed axisymmetric cylinder strains as a function of radius were transformed to principal material coordinates, and input into the nonlinear compressive strength model described above to predict the external pressure at initial material failure assuming that the cylinder response is linearly elastic.

The ABAQUS [8] general purpose finite element code was used to evaluate end closure effects, determine the elastic bifurcation buckling pressure and to evaluate the nonlinear elastic collapse strength of the cylinder. It was assumed that the cylinder is 'long' and, therefore, will buckle with two circumferential lobes. This assumption is supported by closed form energy solution results. Therefore, only a 90° arc segment of the cylinder is necessary to predict the buckling pressure and mode shape. Simple four noded, quadrilateral shell elements which include transverse shear deformation were used in the finite element analyses. A plot of the typical axisymmetric prebuckling displacement pattern and lowest asymmetric buckling mode is shown in figure 3. Although the ABAQUS idealization is relatively simple, the linear axisymmetric displacement results at cylinder midbay compared well with the closed form elasticity solution.

It is well known that linear elastic bifurcation results overpredict the actual collapse load of structures. A large data base of experimental results exists for hydrostatically loaded isotropic cylinders which support this statement. It is generally accepted that small geometric imperfections initiate nonlinear structural behavior, and significantly reduce the cylinder collapse load. This phenomenon can be mathematically modeled in ABAQUS, by imputing a small deviation in the radius as a function of axial and circumferential position and performing a nonlinear incremental loading analysis. As a worst case approach, the initial imperfection pattern was chosen to be the lowest energy mode predicted by the bifurcation buckling solution. Bending strains in the circumferential and axial directions at the inner and outer shell thickness were calculated and input into the nonlinear compressive strength model. The applied bundle (ply) stresses were plotted versus fiber bundle compressive strength to determine the value of pressure at which material failure is initiated.

It should be noted that only initial material failure is predicted. The cylinder may still retain load carrying capacity at a reduced stiffness after initial ply failures. This is analogous to predicting onset of yield in the failure process of a metal structure, while it is known that ultimate failure occurs by forming of a plastic hinge mechanism. A nonlinear material model which includes the inelastic microbuckling strength prediction must be coupled to the structural analysis code to predict the progressive failure of the entire cylinder thickness.

## PREDICTED AND MEASURED CYLINDER FAILURES

The physically based compression strength model and structural analyses methods described above were used to predict the collapse of a series of cylinders tested during the CMFS project. These cylinders were fabricated of all graphite-epoxy, all S2 glass epoxy and hybrids of graphite and glass fiber composite. This section describes analysis results, including predicted mode and location of failure, and compares the analyses to experimental data for two of the cylinders.

### Cylinder HWB1

The cylinder designated HWB1 is a model scale cylinder of the AUSS hull described in reference [9]. The cylinder was designed and wet filament wound at ORNL using IM6 graphite fibers. Approximate fiber distribution was 69% in the circumferential direction and 31% in the axial direction. The cylinder was tested on rigid end supports, and failed catastrophically at 12,500 psi external hydrostatic pressure.

### Stress, Stability and Failure Analysis of HWB1

Calculated strains at 10000 psi external pressure versus the experimental strain gage readings are plotted in figure 4. The elasticity analysis provides a good prediction (within 3%) of the measured circumferential strains at the inner radius. The correlation between the axial strain results is also quite good (within 2%). The strain data correlation suggests that the material properties and structural response of the HWB1 model were characterized to a sufficient degree to predict material failure.

The calculated circumferential, axial, and radial strains at the inner, mid and outer radius were used in the inelastic microbuckling model to determine the pressure at which initial material failure will occur at cylinder midbay. The compressive strength model predicts initial material failure to occur at the inner radius, in the circumferential bundles at an external pressure of 32200 psi. If an inner ply is slightly misaligned with respect to the cylinder tangential direction, initial material failure will occur at a lower value of applied load. Based upon visual observations [10], and the results of recent work on correlation of measured versus experimental data, an initial local fiber waviness of 2° was assumed to be present in some circumferential plies of the cylinder. This reduces the calculated pressure at which microbuckling failure occurs to 20900 psi.

The effects of assumed end conditions on the calculated instability pressure of cylinder HWB1 were evaluated using the ABAQUS finite element code. The effect of neglecting transverse shear deformation (classical plate theory vs. shear deformation theory) in the bending stiffness of the composite shell was also determined. A reduction in

buckling pressure of approximately 15-25% is predicted if the deflection due to shear deformation is included in the plate theory. The effect becomes more pronounced as the fixity of the end conditions is relaxed. The linear finite element analyses predict that the cylinder is elastically stable to 23800 psi if tested on rigid end closures.

A refined elastic collapse strength for the cylinder was predicted by performing a nonlinear incremental loading analysis using large deflection theory. To evaluate the sensitivity of the cylinder to local out of roundness effects, initial imperfection amplitudes of 5 and 50 mils (0.005, 0.050 inches respectively) were input into the ABAQUS finite element model. The fabrication quality of the HWB1 model scale cylinder was reported to be quite good [11]. These amplitudes were used simply to investigate the sensitivity of the predicted failure pressure to slight geometric imperfections.

Results of the nonlinear finite element analyses of the HWB1 cylinder are presented in figure 5. Normal (radial) deflection for two nodes 90 degrees apart on the circumference are plotted versus applied hydrostatic pressure. The elastic collapse strength is defined by the asymptotic increase in normal deflection for a small increase in applied pressure. Also indicated on the figures for comparison is the linear elastic bifurcation buckling pressure. The load deflection curve for the 5 mil initial out of roundness very closely parallels the theoretical Euler type behavior. However, the 50 mil initial out of roundness produces significant nonlinear behavior at much smaller load levels. The elastic nonlinear collapse strength for the 50 mil deviation in radius is predicted to be 17600 psi (approximately a 13% decrease in load).

It has been shown that initial out of roundness patterns inherent to all cylinders produce an asymmetric bending response. Maximum circumferential stresses at the cylinder midbay for the 5 and 50 mil assumed initial imperfections were used in the compressive strength model to predict a nonlinear inelastic type failure (figure 6). This approach predicts that material failure will occur at 19700 psi for the 5 mil imperfection and 14600 psi for the 50 mil imperfection if all plies are perfectly aligned with the cylinder circumference. A 2° circumferential fiber bundle misalignment reduces the predicted pressure at initial microbuckling failure to 17500 psi and 11100 psi, for the assumed out of roundness values of 5 and 50 mils, respectively.

To approximate the rigidity of the end closure structure, strains versus position along the length of the cylinder were computed assuming complete rotational and displacement fixity at the ends. While the circumferential stress near the end closure is reduced, high axial bending and shear stresses will be generated. The maximum calculated axial strain (at the inner radius) is approximately 70% higher than the nominal midbay value. The average thru thickness shear stress in the composite, at the end plate, is calculated to be 0.88 psi per psi of applied pressure. The interlaminar shear strength of the HWB1 material was reported to be 13.0 ksi [9], therefore, shear stresses at the end closures may cause initial interlaminar damage to the shell at

approximately 11000 psi pressure. The compressive strength model predicts initial axial microbuckling failure to occur at 14100 psi external pressure assuming linear behavior to failure. The HWB1 model scale cylinder imploded at 12500 psi external hydrostatic pressure. Strain gage results of the model scale test cylinder showed a significant variation in readings around the circumference [9] indicating some degree of asymmetric behavior. The analyses predict microbuckling failure in the circumferential fibers at the experimental failure pressure, if an initial imperfection amplitude of approximately 35 mils forming two circumferential waves around the cylinder circumference is present. This amount of cylinder out of roundness was not reported.

Based upon the nonlinear collapse analysis results, it is unlikely that structural instability is the initiating failure mechanism for the HWB1 cylinder. Experimental results implied that the graphite epoxy material compressive strength is less than acceptable, since nominal axisymmetric circumferential ply stresses at cylinder failure are calculated to be approximately 90 ksi. It is believed that testing the HWB1 model scale to implosion on rigid end plugs significantly reduced the cylinder strength. The large discontinuity stresses developed due to the radial deflection mismatch at the ends may in fact be the initiating mechanism of failure. Unfortunately, axial strains near the end of the cylinder were not monitored during testing on the rigid end closures.

#### Cylinder SWT1

The cylinder designated SWT1 is a hybrid concept designed and wet filament wound under the ORNL program [12]. Approximate fiber distribution was 50% circumferential IM6 fibers and 50% axial S2 glass fibers. The cylinder was tested on rigid end supports, and delaminated at the inner radius at 16000 psi external pressure.

#### Stress, Stability and Failure Analyses of SWT1

The calculated strains at 10000 psi external pressure versus the experimental strain gage readings for SWT1 are plotted in figure 7. The elasticity analysis provides an excellent prediction of the measured circumferential and axial strains at the inner and outer radii. As before, the calculated were input into the inelastic microbuckling model to predict the pressure at which initial material failure occurs at cylinder midbay. The compressive strength model predicts initial material failure to occur at the inner radius in the graphite fiber circumferential plies at an external pressure of 24600 psi. The calculated pressure required to initiate material failure in a region of 2° fiber misalignment is 16000 psi. This quantitatively illustrates the sensitivity of graphite fiber composite cylinder strengths to slight material imperfections. The axial plies in the SWT1 are S2 glass epoxy, and the calculated failure strength due to axial stresses is virtually unaffected (3% reduction) by a 2° bundle misalignment.

The length to diameter ratio for hydrostatic testing of cylinder SWT1 was 1.5, 25% less than the HWB1 cylinder. The S2 glass axial plies produce composite shear moduli which are higher than HWB1. The combination of a lower length to diameter ratio and higher composite shear modulus increases the calculated buckling strength of this cylinder relative to cylinders HWB1 and RGR1. Assuming fully fixed ends, the elastic buckling prediction for SWT1 is 26200 psi.

The imperfection sensitivity of the SWT1 cylinder was studied in the same manner as the HWB1 cylinder. The assumed 5 and 50 mil out of roundness patterns discussed previously were input into the ABAQUS model and nonlinear elastic collapse pressures were determined. Radial deflection is plotted versus load in figure 8. The nonlinear analysis using the 5 mil imperfection predicts a collapse strength within 4 percent of the bifurcation buckling load (as expected). The collapse prediction is reduced by 15%, to 20300 psi for the 50 mil initial cylinder out of roundness. The relatively high elastic buckling predictions, and knowledge that the compressive strength of graphite fiber composites are extremely sensitive to local material imperfections found in cylindrical geometries, suggested that elastic buckling is not likely to be the initiating failure mechanism for the SWT1 cylinder model.

Figure 9 plots the maximum circumferential stresses at midbay of the SWT1 cylinder versus hydrostatic pressure for an assumed initial cylinder out of roundness of 5 and 50 mils. Computed fiber bundle strengths were plotted versus the applied stresses to predict a nonlinear inelastic type failure. This approach predicts that material failure will occur at 20800 psi for the 5 mil imperfection and 15600 psi for the 50 mil imperfection if all plies are perfectly aligned with the cylinder circumference. A 2° fiber misalignment reduces the predicted pressure at initial microbuckling failure to 16000 psi and 10900 psi for the assumed out of roundness values of 5 and 50 mils, respectively.

The zero rotation and radial displacement constraint imposed by the test configuration produces a large axial bending moment and shear stress in the cylinder. The maximum axial stress near the cylinder end (inner radius) is calculated to be 80 percent higher than at midbay. The average shear stress at the cylinder end in the radial direction is calculated to be 0.49 psi per psi of applied pressure. However, for cylinder SWT1, interlaminar failure is not a concern to pressures well above predicted cylinder failure due to the high shear strength of the S2 glass axial plies.

Figure 10 plots experimental and ABAQUS calculated strains versus pressure on the interior surface for two axial distances from the ends of the cylinder. The significant bending moment (and stress gradient) induced by the fixed ends is clearly illustrated by the large difference in calculated strains over an axial distance of only 1.25 inches. Also, the experimental results appear to closely follow the calculated strains at 1.50 inches in from the cylinder end. Reference [11] indicated that the gages were placed approximately 1.00 inches from the end. Based upon correlation of the experimental versus calculated

strains, it appears that the relatively coarse ABAQUS mesh is accurately predicting the cylinder response under hydrostatic loading. If the results are projected back to 0.25 inches from the cylinder end, the compressive axial strain in the S2 glass bundles at 16000 psi pressure exceeds 1.7 percent.

The pressure at which initial compressive microbuckling failure occurs in the SWT1 cylinder model is not a strong function of the assumed end conditions. However, the end conditions do produce high axial ply stresses at the inner radius which may be a contributing mechanism to the failure process. The ABAQUS analysis combined with the material microbuckling model indicates that the axial plies near the ends should fail at approximately 14500 psi pressure if a slight bundle misalignment is present. The pressure at which failure is initiated is increased to approximately 16500 psi if the bundles are perfectly aligned with the axis of the cylinder. Failure of the circumferential bundles at midbay, using an initial cylinder out of roundness of 5 mils or less, is predicted to initiate at 16000 psi external pressure (20000 psi if zero bundle misalignment is assumed). Analysis results showed that the cylinder was structurally stable to well beyond this load. The testing of SWT1 was halted at 16000 psi external hydrostatic pressure due to high acoustic emission. Axial cracks at the inner radius were observed, indicating circumferential ply failure had occurred.

#### CONCLUDING REMARKS

The initiating mechanism of failure in a unidirectional composite whose primary stress component is compression in the fiber direction is local instability. The nonlinear microbuckling model shows that the local instability compression strength is a function of the shear modulus of the composite. The shear modulus of the composite depends upon the nonlinear shear modulus of the matrix, which can be determined by the effective state of stress in the matrix. Therefore, a key factor in predicting failure of composites in thick walled cylindrical geometries is the accurate determination of the local multidirectional state of stress on the fiber bundle (ply). Stress distributions which may be unimportant to the failure of isotropic materials may significantly reduce the compressive strength of anisotropic composites. This suggests that the compression strength of a fiber composite under multiaxial stress in a thick cylindrical specimen will differ from the strength determined by uniaxially loading a flat coupon. As such, caution must be exercised in the extrapolation of uniaxially loaded flat coupon strength measurements to cylinder strength.

Slight fiber misalignments with respect to the primary load path in a composite leads to a reduction in compression strength with respect to a perfectly oriented material. It is suggested that material failure is initiated in an area of initial ply waviness. The magnitude of strength reduction is dependent upon the applied state of stress and many key material variables. Analytical and experimental



results indicate that composites with high modulus anisotropic fibers in a low modulus resin will be affected most significantly by slight material imperfections. The compression strength of intermediate modulus graphite epoxy materials has been shown to be more sensitive to initial fiber waviness than S2 glass epoxy composites.

An approach for the strength evaluation of unstiffened hydrostatically loaded composite cylinders has been defined which considers both material strength and structural stability modes of failure. The methodology for failure prediction was successfully applied to a series of cylinders tested during the CMFS program. For each of the cylinders studied, small geometric out of roundness was shown to significantly reduce the overall collapse strength. It should be noted that the cylinders were designed based upon nominal stress states, and did not include the discontinuity stresses developed at end closures used for hydrostatic testing. The nonlinear collapse analyses indicate that the testing of cylinders containing graphite epoxy axial plies on flat rigid end closures significantly reduced the overall collapse strength. The rigid end constraints did not significantly reduce the computed and measured collapse strengths of cylinders which contained S2 glass axial reinforcement.

#### ACKNOWLEDGMENT

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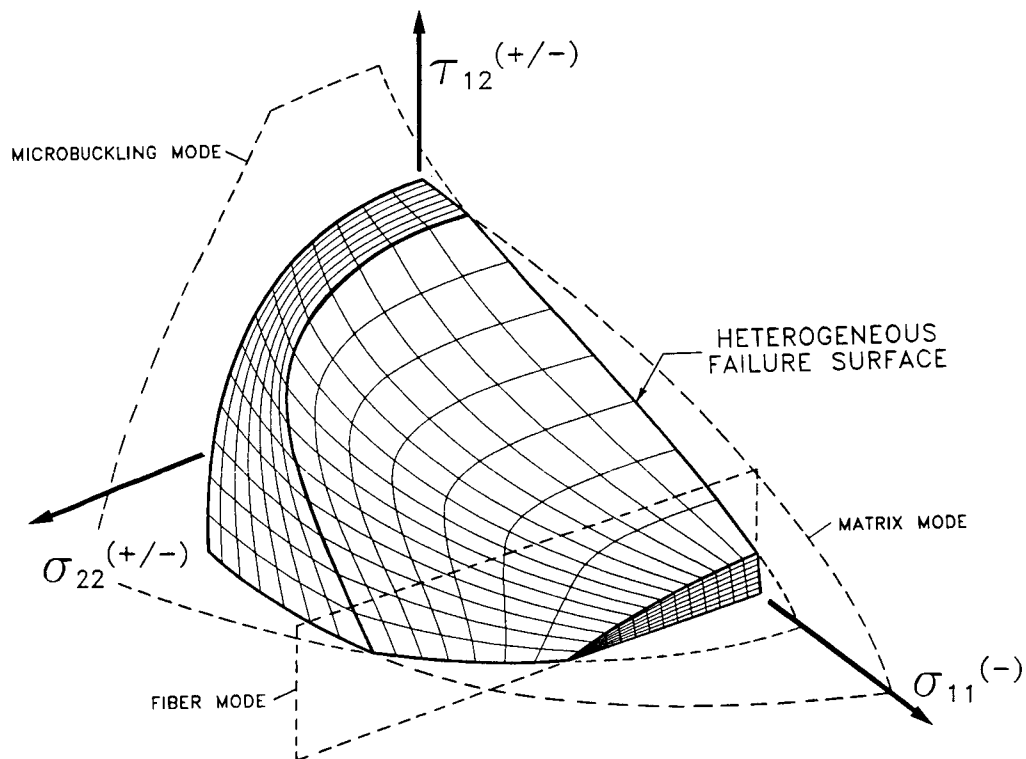


Figure 1. Graphical Representation of Failure Surface Showing Interaction of Modes

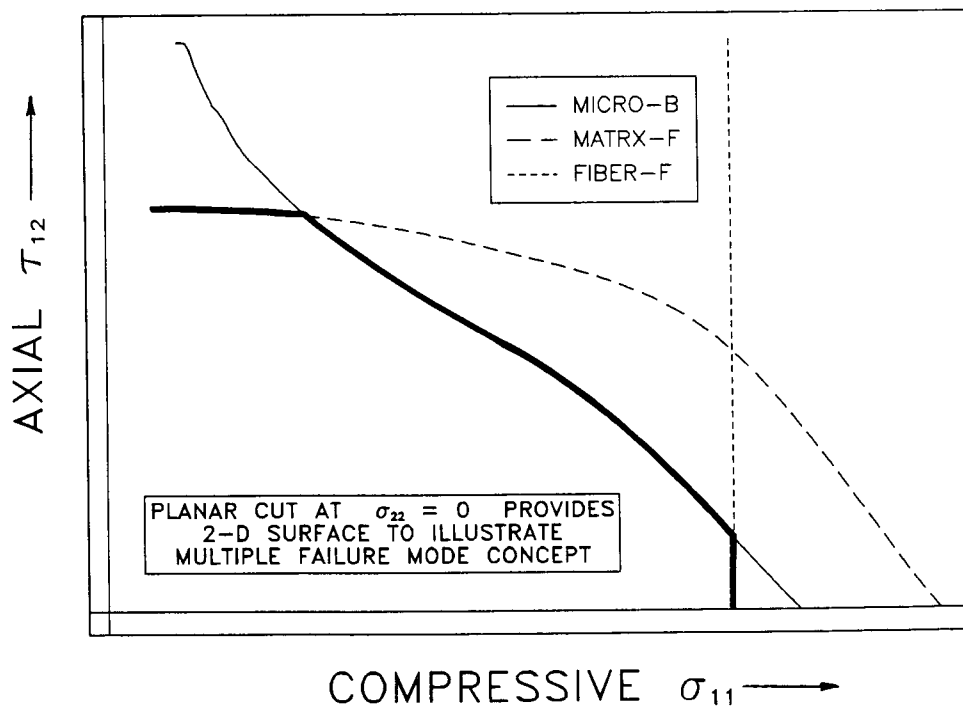
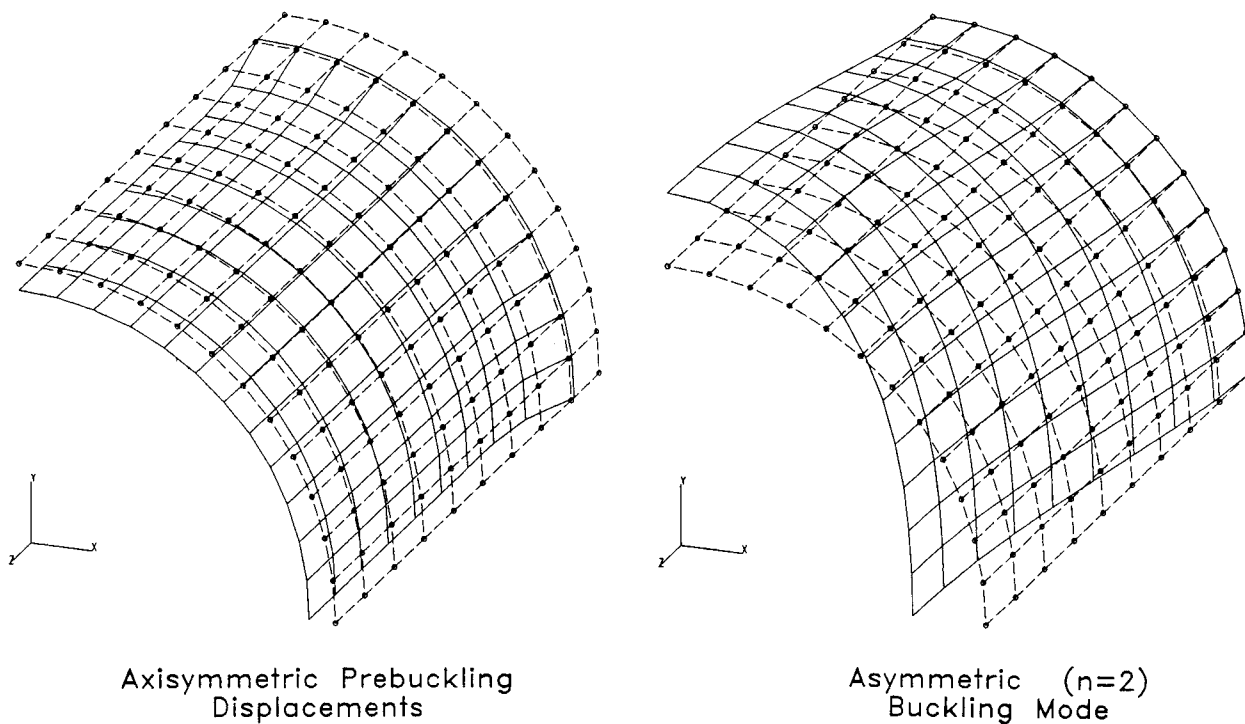
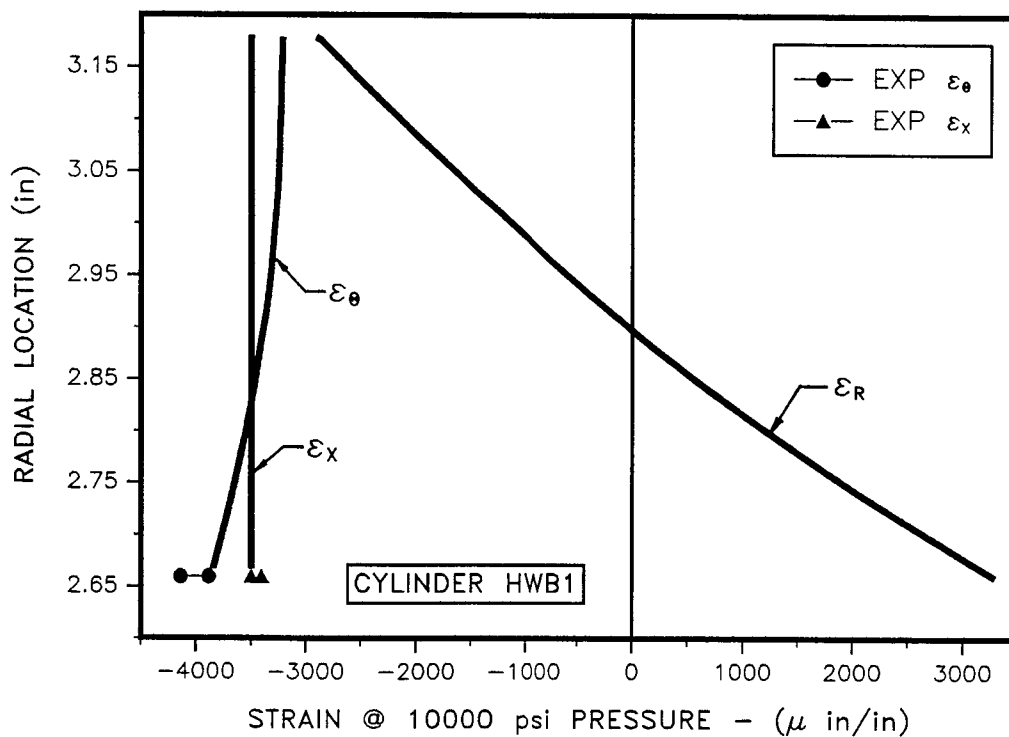


Figure 2. Simplified Failure Surface in Two Dimensional Stress Space



**Figure 3. ABAQUS Idealization Used to Predict Cylinder Response to Hydrostatic Pressure**



**Figure 4. Predicted and Measured Cylinder Strains Versus Radius for HWB1**

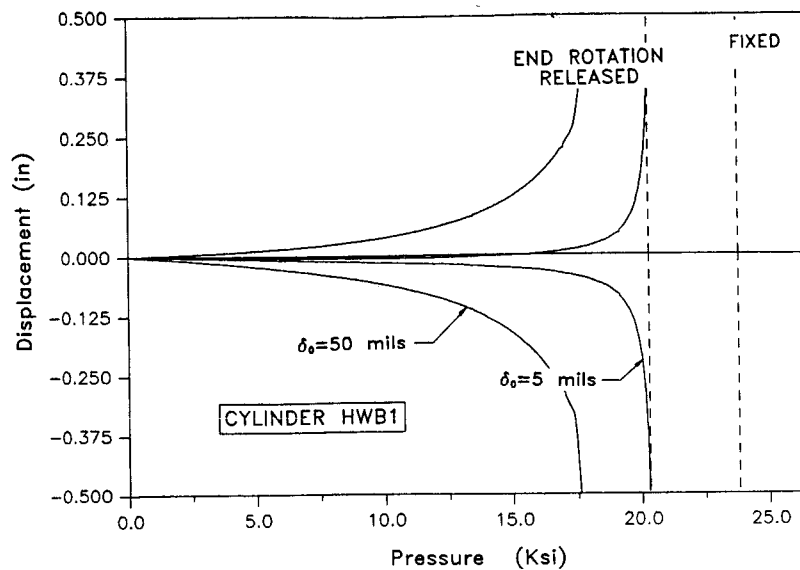


Figure 5. Elastic Nonlinear Collapse Analysis Including Initial Cylinder Out-of-Roundness

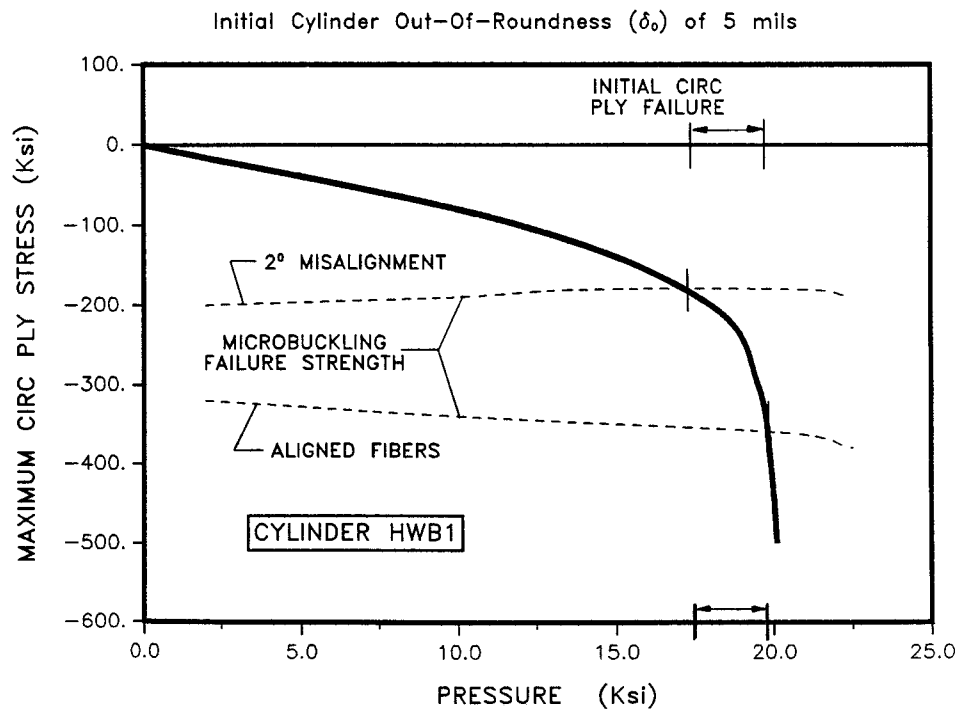


Figure 6. Determination of Initial Material Failure at Midbay of Cylinder HWB1

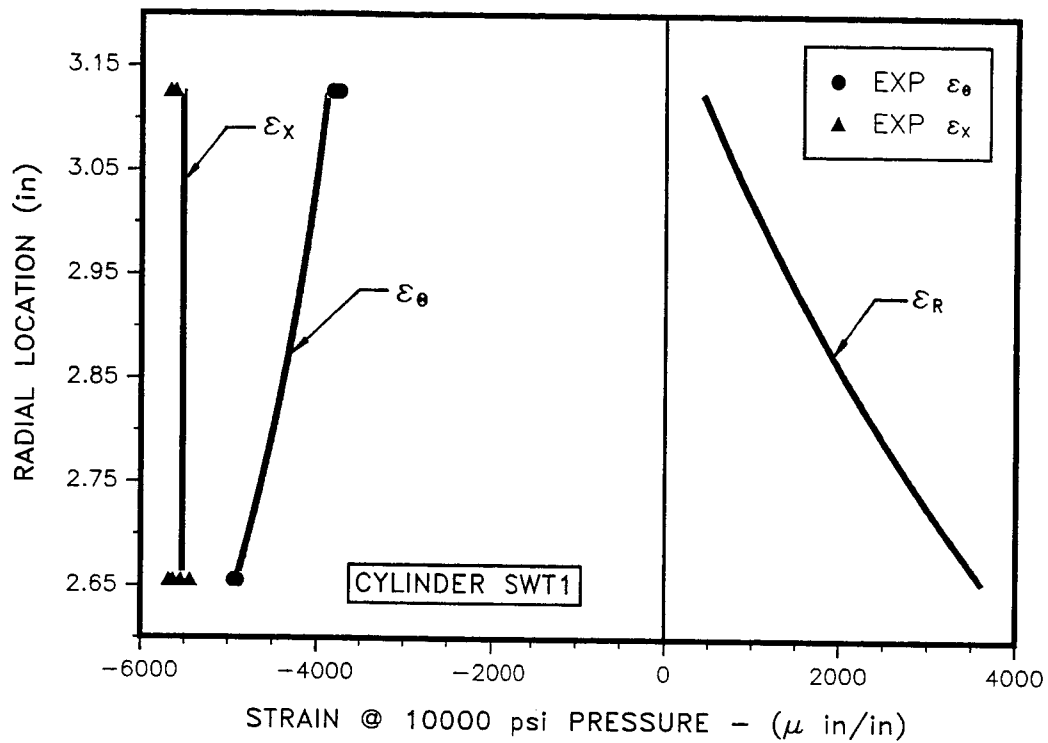


Figure 7. Predicted and Measured Cylinder Strains Versus Radius for SWT1

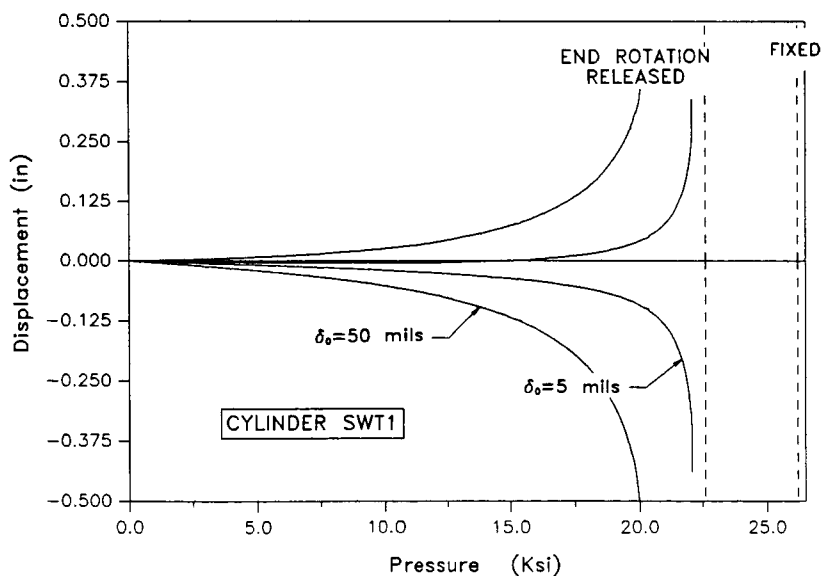


Figure 8. Elastic Nonlinear Collapse Analysis Including Initial Cylinder Out-of-Roundness

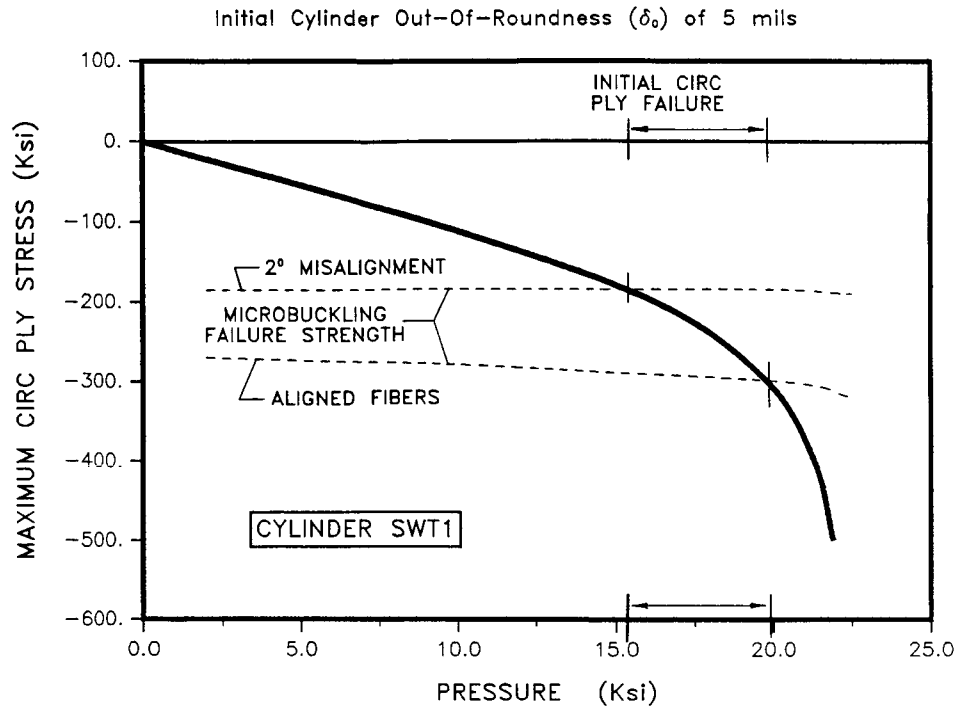


Figure 9. Determination of Initial Material Failure at Midbay of Cylinder SWT1

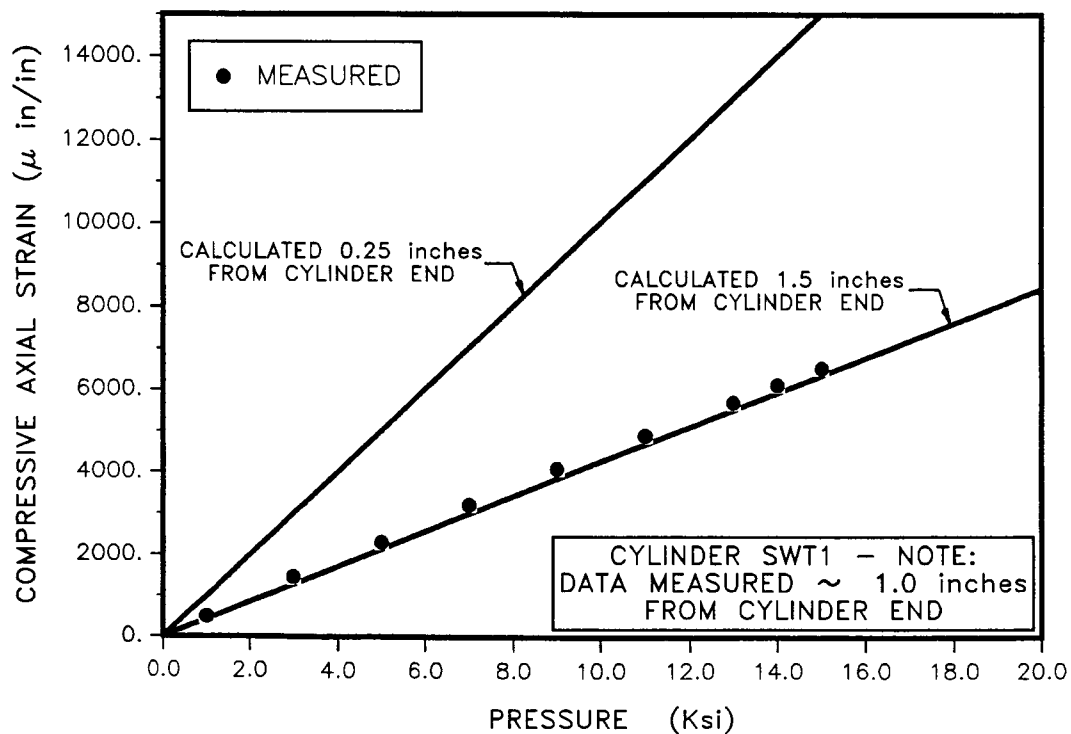


Figure 10. Measured and Predicted Axial Strains at Inside Surface Near End Closure of SWT1

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## ON THE THERMALLY-INDUCED RESIDUAL STRESSES IN THICK FIBER-

## THERMOPLASTIC MATRIX (PEEK) CROSS-PLY LAMINATED PLATES

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## SUMMARY

An analytical method for calculating thermally-induced residual stresses in laminated plates is applied to cross-ply PEEK laminates. We considered three cooling procedures — slow cooling (uniform temperature distribution), convective and radiative cooling, and rapid cooling by quenching (constant surface temperature). Some of the calculated stresses are of sufficient magnitude to effect failure properties such as matrix microcracking.

## INTRODUCTION

Thermally-induced residual stresses in laminated composites are introduced by fabrication and by environmental exposure. They are an unavoidable consequence of (1) the nonuniform distribution of cooling temperature due to the phenomenon of heat transfer and (2) the difference in thermal expansion coefficients of lamina in the fiber direction and the transverse direction. For thin laminated plates the residual stresses caused by (1) may be ignored. But, for thick laminated plates the residual stresses caused by (1) can be as large as those caused by (2). Tensile residual stresses in off-axis plies (*e.g.* 90° plies) are particularly important because they may be large enough to promote damage by matrix microcracking. The prediction and measurement of residual stresses are therefore important topics that are relevant to production, design, and performance of composite components.

Residual (or thermal) stresses and heat transfer are classical problems for conventional materials. A number of investigations specific to composite materials are available (*e.g.* Refs. [1-6]). The theoretical and experimental investigations for residual stresses in Refs. [1-4] and [6] illustrate the residual stresses due to unequal thermal expansion coefficients in the fiber and the transverse directions. Using the finite difference method (for temperature) and finite element method (for thermal stresses), Chen *et al.* [5] studied the failure of laminates under thermal and mechanical loading with the consideration of heat transfer. But only few investigators have considered the residual stresses caused by the nonuniform distribution of temperature, which is especially significant for thick laminates. The goal of present study is to gain insight into the mechanisms of thermally-induced residual stresses in cross-ply laminates, which are caused by both disparate thermal expansion coefficients and by nonuniform distribution of temperature during cooling.

Thermoplastic matrix (PEEK) composites have received much attention due to their high stiffness and high fracture toughness. The stress-free temperature in PEEK composites was measured to be about 310°C [1,6]. The processing temperature, melting temperature, and crystallization temperature are all above 310°C. We therefore treat PEEK as being fully crystallized at the stress-free temperature and calculate the residual stresses that develop on cooling from the stress-free temperature to room temperature. The problem can be separated into two discrete parts. The first part is the analysis of temperature distribution and the second part is the development of residual stresses for given temperature distribution. The problem is separable because heat transfer is not affected by the presence of residual stresses. We use a coordinate system centered inside the cross-ply laminates having the x-axis aligned with the fiber direction of top ply group and the z-axis perpendicular to plane of the plate.

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Our goal is to study the effect of heat transfer on the distribution of the thermally-induced residual stresses. To achieve this goal we consider three cooling procedures: 1) slow cooling in which temperature is uniform over entire thickness and residual stresses are uniform over each laminate group (*i.e.* heat transfer is ignored); 2) cooling under room temperature air by convection and radiation; 3) cooling after the surface is "quenched" to room temperature, (*i.e.* the surface temperature is equal to room temperature). The first and third cases provide two extreme conditions: infinitely large thermal conductivity (slow cooling) and infinitely large convection (or radiation) coefficient (constant surface temperature).

## PART ONE — HEAT TRANSFER

To solve the problem analytically, we use the following assumptions: 1) Heat convection and radiation are assumed to take place only in the thickness direction. Any heat transfer around the edge is neglected. This assumption reduces the analysis to a one-dimensional problem. 2) Although the thermal conductivity,  $k$ , mass density,  $\rho$ , and the specific heat,  $C_p$ , all are functions of temperature, the ratio  $k/\rho C_p$  is assumed to be temperature independent. 3) Linearization of thermal boundary conditions is assumed to be acceptable.

The governing differential equation for heat conduction without any inside heat source (Fourier equation) is [7]

$$\frac{\partial}{\partial z} \left[ k(T) \frac{\partial T}{\partial z} \right] = \rho(T) C_p(T) \frac{\partial T}{\partial t}$$

or when the ratio  $k/\rho C_p$  is independent of temperature

$$\alpha \frac{\partial^2 T}{\partial z^2} = \frac{\partial T}{\partial t} \quad (1)$$

where  $\alpha = k/\rho C_p$ . The analysis of slowly cooled laminates does not involve any heat conduction analysis. The analyses of laminates cooled by convection and radiation or by quenching both use Eq. (1), but require different boundary conditions.

The heat convection and radiation boundary condition has the form

$$-k \frac{\partial T(d,t)}{\partial z} = I_r + I_c \quad (2)$$

where  $d$  is the half thickness of the plate,  $I_r$  and  $I_c$  are the surface energy losses due to radiation and convection, respectively, which are normally assumed to be

$$I_r = \sigma \epsilon [T_{su}^4 - T^4(d,t)] \quad (\text{Stefan-Boltzmann law}) \quad (3)$$

$$I_c = h_c [T_r - T(d,t)] \quad (\text{Newton cooling law}) \quad (4)$$

where  $h_c$  is the convection coefficient;  $T_r$  is the air recovering temperature;  $\sigma$  is the Stefan-Boltzmann constant;  $\epsilon$  is the surface emissivity of a "gray body" (instead of "black body"); and  $T_{su}$  is the temperature of an object surrounding the composite plate and receives the radiated heat. Because  $h_c$  and  $T_r$  are very complex functions of surface temperature,  $T(d,t)$  [5], both the convection and radiation parts of this boundary condition are nonlinear. To solve the problem analytically, we have to "linearize" the boundary condition. We linearize the convection part by assuming  $h_c$  to be temperature independent and letting  $T_r$  equal room temperature —  $T_0$ . We linearize the radiation part by letting  $T_{su} = T_0$  and using the following simplification

$$I_r = \sigma \epsilon [T_0^4 - T^4(d,t)] = \sigma \epsilon [T_0^2 + T^2(d,t)][T_0 + T(d,t)][T_0 - T(d,t)] = h_r [T_0 - T(d,t)] \quad (5)$$

where  $h_r = \sigma \epsilon [T_0^2 + T^2(d,t)][T_0 + T(d,t)]$  is assumed to be approximately independent of temperature [8,9]. Further we let  $T^*(z,t) = T(z,t) - T_0$  and consequently the boundary condition is not only linear but also homogeneous:

$$\frac{\partial T^*(d,t)}{\partial z} = \left(\frac{h_c + h_r}{k}\right) T^*(d,t) = \beta T^*(d,t) \quad (6)$$

where  $\beta = (h_c + h_r)/k$ . Together with the boundary condition at the symmetric axis

$$\frac{\partial T(0,t)}{\partial z} = \frac{\partial T^*(0,t)}{\partial z} = 0 \quad (7)$$

and the initial conditions

$$T(z,0) = T_{sf} \quad \text{and} \quad T^*(z,0) = T_{sf} - T_0 \quad (8)$$

where  $T_{sf}$  is the stress-free temperature, we can solve Eq. (1) by the method of separation of variables.

The general solution takes the form

$$T^*(z,t) = e^{-\lambda^2 \alpha t} (A_1 \sin \lambda z + A_2 \cos \lambda z) \quad (9)$$

Equation (7) yields  $A_1 = 0$  and Eq. (6) gives

$$\lambda \tan \lambda d = \beta$$

This is a characteristic equation having an infinite number of roots —  $\lambda_n$ . Because the differential equation (Eq. (1)) is linear, any possible linear combination of the solutions is also a solution. The general solution then becomes

$$T^*(z,t) = \sum_{n=1}^{\infty} a_n e^{-\lambda_n^2 \alpha t} \cos \lambda_n z \quad (10)$$

The remaining task is to determine  $a_n$  by the initial condition (Eq. (8)). This task can be done analytically only if  $\cos \lambda_n z$  is an orthogonal series, which happens when either  $\cos \lambda_n$  or  $\sin \lambda_n$  is zero for all  $n$  (*i.e.* is  $\lambda_n = n\pi$  or  $\lambda_n = n\pi + \pi/2$ ) and the problem has the appropriate boundary conditions.

Because  $\beta$  is positive, we apprehend that  $\lambda_n \approx (n-1)\pi$  for  $n=2, 3, 4, \dots$  and the larger the  $n$ , the closer they are. Therefore Eq. (10) can be approximately treated as an orthogonal series. By normalizing the half thickness of plate  $d$  to unity and using one of the criteria for the orthogonality condition in Ref. [8], we obtain

$$a_n = \frac{1}{b_n} \int_0^d (T_{sf} - T_0) \cos \lambda_n z \, dz \quad (11)$$

where

$$b_n = \int_0^d \cos^2 \lambda_n z \, dz$$

Substitution of  $b_n$  into Eq. (11) gives

$$a_n = \frac{4(T_{sf} - T_0) \sin \lambda_n d}{2\lambda_n d + \sin 2\lambda_n d} \quad (12)$$

and finally, we have

$$T(z,t) = T_0 + (T_{sf} - T_0) \sum_{n=1}^{\infty} \frac{4 \sin \lambda_n d}{2\lambda_n d + \sin 2\lambda_n d} \cos \lambda_n z e^{-\lambda_n^2 \alpha t} \quad (13)$$

Because the expression for  $a_n$  is approximate, we found it necessary to include more than 100 terms in Eq. (13) to get convergence to the correct answer.

For quenched laminates or laminates with a constant temperature surface, the boundary condition is simply

$$T^*(d,t) = 0$$

The general solution (Eq. (9)), symmetry condition (Eq. (7)) and initial condition (Eq. (8)) are still valid. The above boundary condition reveals

$$\cos \lambda_n d = 0$$

which results in

$$\lambda_n = n\pi + \pi/2 \quad n=1, 2, 3, \dots$$

We can evaluate  $a_n$  by the same procedure used for convection and radiation cooling except that  $\lambda_n$  now defines an exact orthogonal series and the corresponding expression for  $a_n$  is therefore exact instead of approximate. The final expression of temperature distribution in quenched laminates is the same as Eq. (13) except the values of  $\lambda_n$  are changed.

## PART TWO — THERMALLY-INDUCED RESIDUAL STRESSES

The material used in the present study is ICI PEEK/Hercules AS4 carbon fiber prepreg whose thermal expansion coefficient and Young's modulus were provided by ICI Composites. Due to the fiber dominant nature and the temperature insensitivity of the mechanical and thermal properties of carbon fibers, the mechanical and thermal properties in the fiber direction of composites can be assumed to be temperature independent. Experiments show that for this material only a 2.5% error will be introduced by using this assumption [10]. In contrast, the transverse mechanical and thermal properties are temperature dependent. ICI composites supplied experimental results for transverse mechanical and thermal properties from room temperature to the stress-free temperature.

In the analysis of thermally-induced residual stresses in a cross-ply laminate, so-called Classical Lamination Theory is used. Consider a flat plate of uniform thickness with an available temperature distribution  $T(z,t)$ . Classical Lamination Theory gives [11]:

$$\{\sigma\} = [Q](\{\epsilon\} - \{\alpha\}\Delta T(z,t)) \quad (14)$$

$$\{\epsilon\} = \{\epsilon_0\} + z\{\kappa\}$$

where  $[Q]$  is the stiffness matrix,  $\{\epsilon_0\}$  are the strains in the mid-plane of the plate,  $\{\alpha\}$  are the thermal expansion coefficients and  $\{\kappa\}$  are the plate curvatures. Because we deal with symmetric laminates only, their curvatures due to temperature change are zero and therefore

$$\{\sigma\} = [Q](\{\epsilon_0\} - \{\alpha\}\Delta T(z,t)) \quad (15)$$

The residual stresses are zero at the stress-free temperature and start to build up as the laminate cools below this temperature. We assume that below the stress-free temperature the plate is solidified and the displacement along the thickness direction is uniform. Eq. (15) is a general expression for a

temperature-independent material. Because the material we investigate is strongly temperature dependent, however, Eq. (15) has to be modified. If the temperature has an infinitesimal change from  $T$  to  $T+\Delta T$ , the stresses change by

$$\{\Delta\sigma\} = [Q(T)]\{\Delta\epsilon_0\} - [Q(T)]\{\alpha(T)\}\Delta T(z,t) \quad (16)$$

where  $[Q(T)]$  and  $\{\alpha(T)\}$  are the stiffness matrix and thermal expansion coefficient at temperature  $T$ .

We slice the laminate into  $m$  thin layers and assume that each layer is thin enough to ignore the gradients of temperature as well as stress and strain. Because there are no applied forces, the equilibrium equation is taken to be

$$\int_D \sigma_p dz = 0$$

where  $p = x$  or  $y$  and  $D$  is the thickness domain. For a discretized thickness domain we have

$$\sum_{i=1}^m \Delta\sigma_i \Delta z_i = 0 \quad (17)$$

With the substitution of Eq. (16) into Eq. (17) and taking  $\Delta z_i$  as a constant we easily obtain

$$\sum [Q_{11i}(\Delta\epsilon_x - \alpha_{xi}\Delta T) + Q_{12i}(\Delta\epsilon_y - \alpha_{yi}\Delta T)] = 0$$

$$\sum [Q_{12i}(\Delta\epsilon_x - \alpha_{xi}\Delta T) + Q_{22i}(\Delta\epsilon_y - \alpha_{yi}\Delta T)] = 0$$

where  $Q_{11i}$ ,  $Q_{12i}$ ,  $\alpha_{xi}$ , etc. are elements of the stiffness matrix and of the thermal expansion coefficient vector at temperature  $T$ ; and  $\Delta\epsilon_x$  and  $\Delta\epsilon_y$  are variations of total strains in  $x$  and  $y$  directions, which are  $z$  independent, due to temperature variation from  $T$  to  $T + \Delta T$ . These two equations lead to the expressions for  $\Delta\epsilon_x$  and  $\Delta\epsilon_y$

$$\Delta\epsilon_x = \frac{(\sum Q_{11i} \alpha_{xi} \Delta T + \sum Q_{12i} \alpha_{yi} \Delta T) \sum Q_{22i} - (\sum Q_{12i} \alpha_{xi} \Delta T + \sum Q_{22i} \alpha_{yi} \Delta T) \sum Q_{12i}}{\sum Q_{11i} \sum Q_{22i} - (\sum Q_{12i})^2}$$

$$\Delta\epsilon_y = \frac{(\sum Q_{12i} \alpha_{xi} \Delta T + \sum Q_{22i} \alpha_{yi} \Delta T) \sum Q_{11i} - (\sum Q_{11i} \alpha_{xi} \Delta T + \sum Q_{12i} \alpha_{yi} \Delta T) \sum Q_{12i}}{\sum Q_{11i} \sum Q_{22i} - (\sum Q_{12i})^2}$$

and eventually two expressions for the residual stress variation of each slice:

$$\Delta\sigma_{xi} = Q_{11i}(\Delta\epsilon_x - \alpha_{xi}\Delta T) + Q_{12i}(\Delta\epsilon_y - \alpha_{yi}\Delta T)$$

$$\Delta\sigma_{yi} = Q_{12i}(\Delta\epsilon_x - \alpha_{xi}\Delta T) + Q_{22i}(\Delta\epsilon_y - \alpha_{yi}\Delta T)$$

Finally the total residual stresses for each slice can be determined by summing each variation of stress in the temperature (or time) domain

$$\sigma_{xi} = \sum \Delta\sigma_{xi}$$

$$\sigma_{yi} = \sum \Delta\sigma_{yi}$$

## NUMERICAL STUDY AND CONCLUSIONS

According to data provided by ICI Composites, we selected thermal conductivity  $k = 0.25$  W/m $\cdot$ °K, specific heat capacity  $C_p = 1.5$  kJ/kg $\cdot$ °K = 0.4167 W-hr/kg $\cdot$ °K, and average density  $\rho = 1.3$  g/cm $^3$ . From other sources we selected the convective heat transfer coefficient  $h_c = 2.5$  Btu/hr ft $^2$  $\cdot$ °F = 14.186 W/m $^2$  $\cdot$ °K [7], Stefan-Boltzmann constant  $\sigma = 0.1714 \times 10^{-8}$  Btu/hr-ft $^2$  $\cdot$ °R $^4$  =  $5.669 \times 10^{-4}$  W/m $^2$  $\cdot$ °K $^4$  [7], and surface emissivity  $\epsilon = 0.92$  [5].

Figures 1 and 2 illustrate the distributions of x-axis residual stresses for thin laminates [0 $_2$ /90 $_2$ ] $_s$  and [90 $_2$ /0 $_2$ ] $_s$ . Figures 3 and 4 show the results for thick laminates [0 $_90$ /90 $_5$ ] $_s$  and [90 $_5$ /0 $_90$ ] $_s$  (about one inch thick). We draw the following conclusions:

- (1) For thin laminates the slow cooling results are close to the convection and radiation results. This implies that the assumption of a uniform temperature distribution is adequate for thin laminates. For thick laminates the convection and radiation results are between those of slow cooling and quenching results, which indicates that an assumption of uniform temperature distribution is not adequate. An accurate estimation of the non-uniform residual stresses in thick laminates must use an analysis that accounts for heat conduction similar to the one in this paper.
- (2) The residual stresses in quenched laminates, as well as in thick laminates under convective and radiative cooling, always have a high gradient at the laminate surface. The magnitude of stress variation in this area remains unchanged regardless of the laminate thickness. The high normal residual stress gradient caused by nonuniform cooling will likely produce a high shear stress gradient, which might cause local delamination.
- (3) Beyond this high gradient stress area the residual stresses within each ply group remain nearly uniform. Thus, the residual stresses away from this area in thick laminates will be nearly unaffected by processing conditions.

## COMMENTS

The processing conditions affect the cooling temperature distribution and may consequently cause nonuniform residual stresses. The effects of processing conditions are strongest near the surfaces of laminate where the residual stresses can differ significantly from those calculated by a simple laminated plate theory that assumes a uniform temperature distribution during cooling. At the center of laminates, a simple uniform temperature distribution gives a good estimate of the ply residual stresses.

Numerical results support the claim that the residual stresses in "quenched" laminates and in slowly cooled laminates provide upper and lower bounds to the residual stresses. Either one may provide the upper bound somewhere and the lower bound somewhere else. The results for convection and radiation, however, will definitely be bounded by the upper and lower bounds. In real applications, it will be very hard, if not impossible, to precisely describe the cooling boundary conditions in processing. If the upper and lower bounds are given, it will help the designer to have an estimate on the level of the residual stresses.

We believe that below the stress-free temperature of PEEK composites that the plate is solidified and is also fully crystallized [6]. Any significant crystallization happening below the stress-free temperature might cause volume reduction and result in extra residual stresses. We also assumed that viscoelastic behavior of PEEK material does not significantly influence the residual stresses. If a great amount of time is spent above the glass transition temperature, it is possible that stress-relaxation will reduce the level of residual stresses. Most residual stresses form, however, when the matrix is stiff and below the glass transition temperature. At these lower temperatures, stress relaxation effects are probably minimal.

## ACKNOWLEDGMENTS

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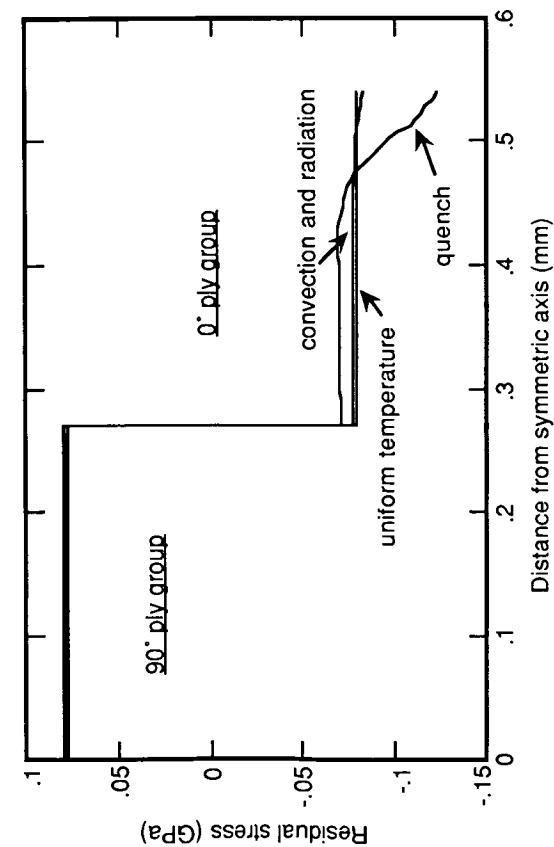


Figure 1: The distributions of x-axis residual stresses in a  $[0_2/90_2]_s$  laminate cooled under uniform temperature distribution (slow cooling), by convection and radiation, and by quenching to room temperature.

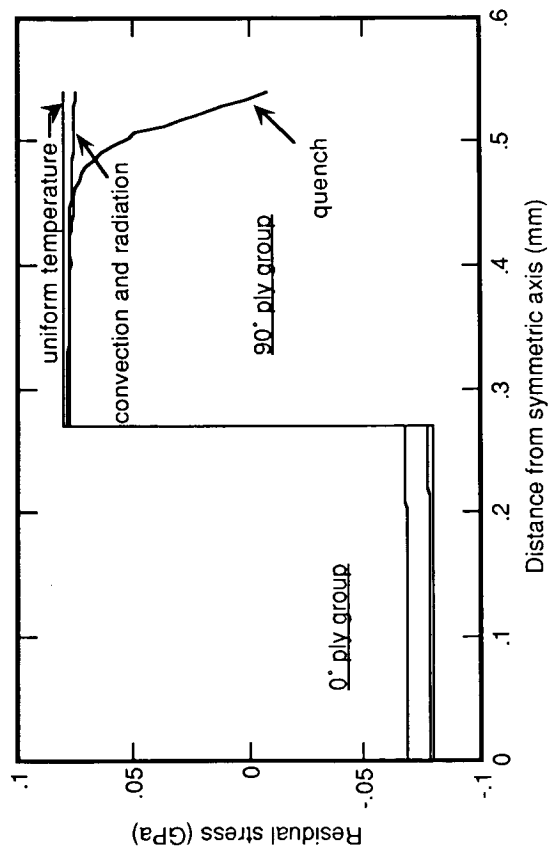


Figure 2: The distributions of x-axis residual stresses in a  $[90_2/0_2]_s$  laminate cooled under uniform temperature distribution (slow cooling), by convection and radiation, and by quenching to room temperature.

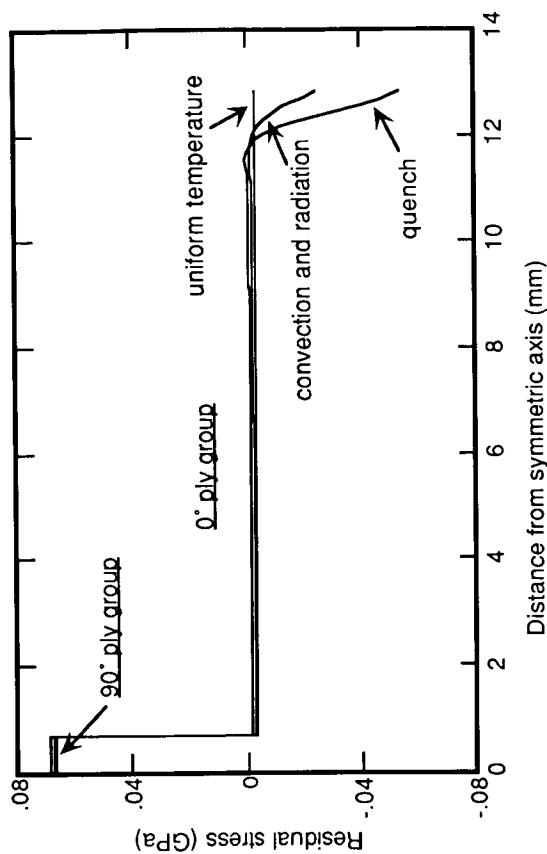


Figure 3: The distributions of x-axis residual stresses in a  $[0_0/90_0]_s$  laminate cooled under uniform temperature distribution (slow cooling), by convection and radiation, and by quenching to room temperature.

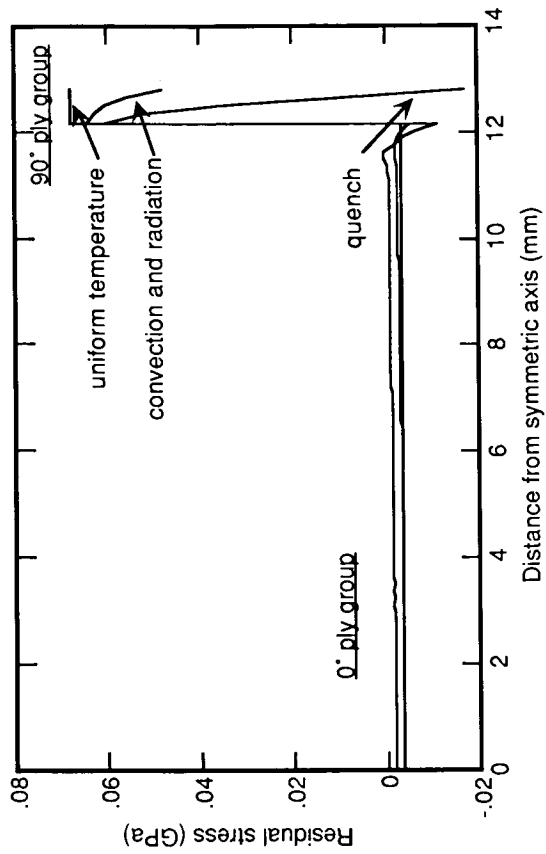


Figure 4: The distributions of x-axis residual stresses in a  $[90_0/0_0]_s$  laminate cooled under uniform temperature distribution (slow cooling), by convection and radiation, and by quenching to room temperature.

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**Advanced Fabrication Processes for  
Thick Composite Submarine Structure**

**James J. Kelly  
DARPA**

**George Leon  
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**Diana Holdinghausen  
McDonnell Aircraft Corporation**



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## LOW - COST DESIGN AND FABRICATION OF COMPOSITE SHIP STRUCTURES

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### SUMMARY

The U.S. Navy has demonstrated a low-cost resin transfer molding process for the fabrication of high performance composite ship structures including monocoque, single skin stiffened, and sandwich configurations. The resulting mechanical properties for the structures are competitive with properties achieved using wet lay-up or prepreg and autoclave cure. Composite design concepts and prototypes have been developed for composite deckhouse, mast, and foundation structures and fabricated using this process. Prototype structural performance has been successfully demonstrated under dynamic load requirements including air blast and shock. The paper also discusses some recent successful experience with the repair of a thick glass-reinforced plastic sonar dome using this process. Some potential near-term applications of the fabrication process for composite ship structures are identified.

### INTRODUCTION

The naval ship structure community is pursuing the challenge of improving the structural performance of naval ships while reducing ship construction and life cycle maintenance costs. In addition to reducing structural weight, efforts are being made to reduce fatigue cracking, corrosion, and noise transmission, increase fragmentation and shock resistance, and improve fire containment. Composite structures offer the clear potential for achieving improvements in all of these areas. Some structural applications where these improvements are under development or consideration are shown in figures 1 and 2.

In order to exploit the advantages of composites for these applications, innovative design concepts, and low-cost processes to fabricate high quality composite structures are needed. In particular, a fabrication process is needed that approaches the cost of glass reinforced plastic (GRP) marine construction (reference 1) but results in the quality of typical aerospace construction. Furthermore, a versatile process is needed that allows for the fabrication of a range of composite structures from thick monocoque to single skin stiffened and sandwich construction using a variety of resins and fiber reinforcements.

The U.S. Navy has been successful in demonstrating a low-cost vacuum resin transfer molding (RTM) process having the capability and versatility just described. The process has been used to fabricate composite deckhouse structural modules, a one - half scale composite mast, and a one - half scale composite foundation. This paper will describe the U. S. Navy's experience with this process and with the

design, fabrication and testing of these composite prototypes. Some limited experience with the use of the vacuum RTM process for repair will also be covered along with some potential near-term applications of this process for the fabrication of affordable high quality composite ship structures.

#### LOW-COST FABRICATION OPTIONS

A number of low-cost options are available for the fabrication of composite ship structures. The most promising of these are resin transfer molding, pultrusion and filament winding. The particular method chosen for a given application depends upon the requirements of the application such as structure or component size, geometry, and desired mechanical properties. Some typical ship structural components would include GRP beams, panels of monocoque, stiffened and sandwich construction, and rectangular and cylindrical modules of similar construction.

Although pultrusion is an attractive process for producing low - cost beams and panels having a uniform or prismatic cross-section, the process is not able to deal with non-uniform or tapered members. While filament winding does not have this limitation, it is generally not able to handle complex structural geometries and the incorporation of stiffeners.

RTM methods, on the other hand, have the versatility to deal with a variety of structural configurations. As an example, they are able to accommodate tapered beams and panels and three dimensional structures having rather complex geometries. Reference 2 is an excellent relatively recent article describing the status of development of RTM methods for advanced composites.

In general, RTM methods seem to fall into one of the three following categories:

- Conventional RTM
- Vacuum Assisted RTM
- Vacuum RTM

With the conventional RTM method, dry fiber reinforcement or preform is loaded into a split mold (usually female and male), the mold is clamped together and resin is injected under pressure. This process is used for the manufacture of such medium to large size parts as car bodies, bus shelters, hatch covers, and shower enclosures (reference 3). The term vacuum assisted RTM has been used in industry to refer to two types of RTM processes, one in which the vacuum assists the process of resin injection 'under pressure', and the second where the resin is injected 'under vacuum' alone. For example, vacuum assistance is often used with conventional RTM to remove air and volatiles from the part during resin injection under pressure. Vacuum assisted has also been used to describe several processes developed in Europe which utilize only a vacuum and no pressure to transfer the resin. One of these has been used to fabricate a 22 m landing craft ( reference 4). The term Vacuum RTM has been introduced in this paper to try to eliminate the confusion and to describe processes where a full vacuum, without pressure, is used to consolidate the dry fiber reinforcement and to inject resin into the structure.

Vacuum RTM, as defined here, is apparently employed at relatively few companies worldwide. Some of the companies either investigating or using vacuum and vacuum assisted RTM processes are listed in Table 1 along with their applications to date. It should be noted that these applications involve the use of an open mold, either male or female, or a closed or split mold. The open mold approach is less expensive because it minimizes the amount of tooling required. It may be used when only a single mold surface is required.

The U.S. Navy has been very successful in evaluating and demonstrating a low-cost vacuum RTM process for the fabrication of composite deckhouse, mast, and foundation structures. The particular vacuum RTM process evaluated is the Seemann Composites Injection Molding Process (SCRIMP). The SCRIMP process uses a single open mold and a patented resin distribution system, including a patented distribution medium, for achieving aerospace quality composites. The patented process has resulted in high fiber content, approximately 70% by weight for glass, and very low void content (<1%). Resultant mechanical properties are considerably greater than properties achieved with standard 50 to 60 % glass content panels and are competitive with properties obtained using wet lay-up or prepreg and autoclave cure procedures (See Table 2).

#### PANEL FABRICATION USING 'SCRIMP'

The first panel fabricated using the SCRIMP process for the U.S. Navy was a 8'x16' hat-stiffened deckhouse panel in figure 3. At the time of its fabrication in 1989 the panel was apparently the largest high performance composite panel fabricated using an RTM process in North America. The SCRIMP process was used to demonstrate a more affordable alternative to the wet lay-up autoclave cure process utilized to fabricate the original panel. The deckhouse panel in figure 3 involves tapered stiffeners, 6" deep and 6" wide at midspan, with skin thickness ranging from 0.5" between stiffeners at midspan to 0.75" at the panel edges. Stiffener flange and web thicknesses varied from 0.2" at midspan to 0.4" at the stiffener extremities. Although the original autoclave panel was fabricated using E- glass and isophthalic polyester resin, the SCRIMP fabricated panel utilized a vinyl ester resin to improve the toughness of the panel under dynamic loading and to facilitate the SCRIMP process.

Figure 4 shows a schematic layout for the SCRIMP process for a segment of the hat stiffened panel. The basic steps in the SCRIMP process for the hat stiffened panel are summarized below.

1. The dry reinforcement preform, including the stiffener cores, are laid up on the rigid mold.
2. The preform is covered by a peel ply, distribution medium, and a vacuum bag in that order.
3. A full vacuum is drawn to consolidate the panel and inject the resin.
4. Resin is allowed to flow into the panel until full wetting occurs.
5. The panel is allowed to cure for approximately 8 hours at room

temperature.

6. The panel is oven post-cured at an average temperature of 140 degrees F for 8 hours.

The most labor intensive part of the process is the laying up of the dry preform for the panel. The base skin (0.32") of the panel, under the stiffener cores, was laid up first out of 8 plies of 50 oz, twill weave fabric, woven by Seemann Composites. Five plies of 50 oz twill weave were then draped over the stiffener cores. Five blocks of 24 oz woven roving at 45° and 90° were feathered into the 50 oz twill weave over the taper region of the stiffeners. The last two columns of Table 2 compare the mechanical properties achieved for the SCRIMP panel with those obtained for the autoclave panel. The properties for the two panels are seen to be comparable with the interlaminar shear strength seen to be clearly superior as expected for the vinyl ester over the polyester panel.

In addition to the hat stiffened panels, a series of 4'x4' and 8'x16' composite sandwich panels were also fabricated using the SCRIMP process and a variety of resin and core materials. The above process, with some variations, was used for all of these panels.

Sandwich panels with foam and balsa cores, up to 4.0 " thick, were laid up and resin injected in one step. The process took advantage of narrow vertical slits in the balsa core product for transferring resin from the upper to lower skin of the panel. Table 2 compares the mechanical properties of the 70 % glass content skins (see columns 2 and 3 ), for some of these sandwich panels, with corresponding 50% glass content skin properties (column 1) resulting from panels produced using standard wet lay-up boat-building methods. The advantage of the SCRIMP process in providing 70% glass content and significantly better mechanical properties is evident.

## DESIGN CONCEPTS AND PROTOTYPE DEMONSTRATIONS

The U.S. Navy has successfully developed and demonstrated composite design concepts for a number of ship structural applications including deckhouses, masts, and foundations. Each of these composite prototypes were fabricated using the SCRIMP process and will be taken up next in the paper.

### Composite Deckhouse

Composites offer a number of advantages for deckhouse application, the principal ones being reduced weight by up to 45%, elimination of corrosion and fatigue cracking, and the capacity for fire containment.

Composite deckhouses must be designed for a number of loadings including air blast, hydrostatic pressure, and hull-deckhouse interaction loads. Since air blast is the governing load condition, extensive dynamic tests have been conducted to evaluate the structural resistance of composite beams, panels, and modules to air blast loading.

The currently favored approach for the shipyard construction of composite deckhouses is the prefabrication of large high quality composite panels and their attachment to an erected supporting steel

framework as indicated in figure 5 (reference 5). Since these panels could be fabricated off-line at a vendor or using a licensed process in a shipyard, the opportunity exists to fabricate the composite panels with extremely low void content and significantly higher mechanical properties than normally achieved with traditional wet lay-up room temperature cure methods used to fabricate large GRP structures such as naval minesweepers and minehunters. Some of current bolted - bonded joint configurations for the connection of composite panels to the steel framework of a deckhouse are shown in figure 6.

Two composite panel design concepts, single skin hat stiffened and sandwich core panels, have been developed for deckhouse and superstructure construction. The use of monocoque GRP panels on closely-spaced steel framing, a solution under development by the British Navy( reference 6), has not been pursued because it is significantly heavier than the use of single skin stiffened and sandwich panels on more widely-spaced framing.

### Single Skin Construction

GRP single skin stiffened construction was initially pursued for deckhouse structure because it involved less risks than sandwich construction in areas such as fatigue and shock resistance, environmental effects and general ruggedness. A tapered hat stiffened panel concept was developed as a basic building block for deckhouse construction. The stiffeners are tapered to make them peel - resistant under loading, to minimize panel weight, and to simplify the joining of panels to steel frames. Joining is facilitated because of the single skin thickness around the perimeter of the panel. The fabrication of the hat stiffened panels using the SCRIMP process was discussed in detail earlier.

In order to evaluate the structural performance of the hat stiffened concept under air blast loading, an 8'x15' panel was fabricated using a wet lay- up autoclave cure procedure. Panels were also fabricated using the SCRIMP process and used in the construction of a 16'x8'x8' module as shown in figure 7(a). The module was assembled using GRP connection angles since GRP was initially favored over steel for the supporting framework of deckhouses and superstructures. Figure 7(b) shows a photograph of the bolted- bonded joint details for this module. Many bolts were used to provide a conservative design.

The module was successfully air blast tested at the White Sands Missile Range under the MISERS GOLD and DISTANT IMAGE events in 1989 and 1991. In the first test, at an equivalent static pressure of approximately 30 psi, the front of the module deflected approximately 2 inches and half moon shaped delaminations appeared along the inside bottom of the front panel where the GRP panel was bolted to a rather rigid steel coaming. The extent of delamination damage (figure 8) grew considerably in the second test, at an equivalent static pressure of a little over 50 psi (maximum front panel deflection of 5.6 inches), but the damage was still acceptable since no rupture of the module panels occurred. The acceptable damage to the module, under the severe dynamic loading, demonstrates the superior toughness of the structure resulting from the SCRIMP process and the vinyl ester resin even at high glass contents ( approximately 70%).

Additional air blast tests have been conducted on one-half scale GRP hat stiffened panels to evaluate and screen a variety of bolted- bonded joint details for use with these panels. These panels, fabricated using the SCRIMP process, were dynamically tested in a blast tube at the Defence Research Establishment Suffield (DRES), Alberta under a collaborative effort with the Canadian Navy. These tests explored the options of greater bolt spacings, the use of countersunk bolts, and the performance of steel versus GRP connection angles. This test series resulted in the streamlined bolted- bonded detail in figure 7 (c) which was adopted in the construction of a composite sandwich module to be discussed next. Figure 9 shows the delamination damage observed after the blast tube test of one of the half scale panels. It was found to be similar to the delamination shown in figure 8 at the bottom of the hat stiffened module for a similar pressure.

This paper has not addressed the integration of electromagnetic issues such as electromagnetic shielding and lightning strike protection into the design. The effectiveness of the SCRIMP process in fabricating composite deckhouse panels with these features must still be demonstrated.

### Sandwich Construction

Sandwich construction offers a number of advantages over single skin stiffened for marine as well as aerospace applications. Some of the particular advantages for naval deckhouses are increased weight reduction over single skin stiffened of 10% or more, an inherent thermal insulation capability, and a flat surface on panel interiors for ease of outfitting.

In order to select the most promising sandwich options for deckhouse structure, the dynamic structural performance of a variety of composite sandwich panels were air blast tested and evaluated in a blast tube at DRES under a joint effort with the Canadian Navy. Various panel geometries (uniform and tapered), cores (balsa, poly vinyl chloride, and polymethacrylimide), resins (vinyl ester, polyester, and phenolic) and glass contents (50% and 70%) were evaluated in the test series. As a result of these tests, a tapered panel fabricated using the SCRIMP process with a balsa core and vinyl ester resin was selected. This panel showed no visible signs of delamination even after air blast testing to a maximum equivalent static pressure of 72 psi.

Following the successful test of the tapered vinyl ester panel, a fire hardened concept was developed by introducing a structural skin at mid-depth in the balsa core panel. This panel withstood a pressure of 155 psi without rupture. The fire hardened tapered panel was successfully manufactured using the SCRIMP process in a two step process. The lower half of the panel was first fabricated by 'scrimping' the middle skin and bottom skin to the lower balsa layer. The upper skin and upper balsa layer was then 'scrimped' to the lower layer.

The above process was used to fabricate the 16'x8' and 8'x8' fire hardened panels for a 16'x8'x8' sandwich module. These panels were bolted and bonded to an erected steel framework comprised of steel angles. The particular epoxy bonding adhesive selected provided good toughness and elongation characteristics under dynamic loading. The module was air blast tested at White Sands with an equivalent static

pressure of approximately 30 psi on the front of the module.

This resulted in a maximum deflection of 4.8 inches of the front of the module and some acceptable delamination in the panel taper around the perimeter of the front panel. The extent of delamination at the inside bottom of the front panel was probably minimized by allowing the steel coaming to deform plastically under loading. A comparison of overall structural performance of the GRP stiffened and sandwich module is given in Table 3.

A comparison of weight and costs to fabricate the individual panels for the stiffened and sandwich modules and to construct these modules is given in Table 4. It should be noted that the fire - hardened sandwich panels in Table 4 were more expensive to fabricate than non fire - hardened panels. Also, these costs are associated with the fabrication of one-of-a-kind prototypes and involve significant engineering and manufacturing development costs. A substantial reduction in costs approaching 50 % is expected for full production. As an example, it is expected that the cost to fabricate the stiffened panels will drop from \$9.40 per pound for the prototype to \$5 to \$8 per pound in production.

### Composite Mast

The weight and height of masts have a significant impact on the stability and seakeeping characteristics of naval ships. Composite masts are therefore a most attractive option for either upgrading the stability of existing ships through backfit applications or providing a greater topside weight growth margin in the construction of new ships. Design studies have shown that a composite mast can be expected to save from 20% to 50% over a conventional metallic mast. In addition to the weight reduction benefits, the use of composites would also eliminate corrosion, improve fatigue performance, and improve the performance of mast sensors by reducing electrical blockage over that of metallic masts.

In order to demonstrate the feasibility of a composite mast for naval combatants, a one- half scale, 36 foot tall, prototype mast( see figure 10) was designed, fabricated and tested under a Cooperative Research and Development Agreement (CRDA) between the David Taylor Research Center (DTRC) and Ingalls Shipbuilding. In developing the prototype mast, the external configuration and geometry of the existing DDG- 51 mast was adopted as a baseline. A weight reduction of 20% was realized against this aluminum baseline design.

The mast structure and elements were designed to meet the vibration requirements ( bending frequency > 3.5 hz.), to have a factor of safety of 1.0 for air blast loads, and to match the ballistic performance of the existing aluminum mast. A hybrid material system of S-2 glass and carbon in a vinyl ester matrix was selected for the main trunk of the mast; S-2 glass was chosen to maximize the ballistic performance and carbon in a +/- 45 degree layup to provide sufficient torsional stiffness. S-2 glass was used for the other elements of the mast.

The composite mast model was constructed by fabricating the main individual elements of the mast, that is, the trunk, two yardarms, and two stays, using the SCRIMP process. These large elements were then shipped to White Sands where they were bolted together and the mast



erected for the DISTANT IMAGE air blast test in June 1991. In the interest of enhancing producibility, commercially available pultruded GRP struts were used to support the yardarms from the trunk and aluminum connection details were used at the extremities of the struts.

Figure 11 illustrates the materials, geometry and details for the fabrication of the main trunk of the mast. The main trunk, 28.5 feet long with a square cross section measuring 27 inches on a side, was fabricated in two halves. Each half of the trunk was a large angle member that was laid-up using dry E-glass and carbon fabric in a female mold and then 'scrimped' in one step. E-glass was used because of the unavailability of S-2 glass to meet the fabrication schedule. An adjustable female mold was used to allow for the effect of anticipated shrinkage on the corner angles. The carbon fabric was imbedded at mid depth in the E-glass laminate to minimize the release of carbon fiber in an explosion. The SCRIMP process was also used to fabricate the corner elements. The entire trunk was then assembled together using limited bolts and an epoxy adhesive that proved effective for the deckhouse module. A similar process was used to fabricate the yardarms and aft legs or stays whose cross-sections are shown in figure 12. The composite mast prototype was fabricated for a cost of \$42 per pound. Since there was considerable engineering and manufacturing development included in this cost, it is felt that composite masts could be produced in full-scale production at a cost of \$20 to \$30 per pound.

The one-half scale mast was tested at the DISTANT IMAGE event to a substantial air blast loading which produced an overall deflection of approximately 6 inches at the top of the mast. Although the dynamic high speed photos showed the mast trunk, and especially the yardarms, to be clearly vibrating from the effects of the air blast, virtually no damage occurred. A careful examination of the mast exterior revealed only one small delamination 2 to 3 inches in diameter on a large connecting bracket for the mast platform. Close agreement was obtained between the measured overall bending frequency for the mast (6.7 hz - one-half scale; 3.35 hz - full scale) and the design value (3.5 hz).

### Composite Foundations

The use of composite materials for foundation structures offers many potential advantages over current steel structures. Weight reductions of over 50% are possible in some applications. Even more important in many naval applications is the potential for improving the noise transmission characteristics of machinery/ equipment foundations. Additional benefits include the mitigation of shock loads to the supported equipment, improved corrosion resistance, and reduced magnetic signature.

In order to demonstrate the structural feasibility of composite materials in foundation structures, a one-half scale pump foundation was designed, fabricated, and tested under an exploratory development program for submarine structures. The baseline selected for this demonstration was a fresh water pump foundation. This foundation is attached to the upper level deck in an engine room and supports a bedplate, pump, and motor. In this demonstration, a composite foundation was designed to match the existing geometrical constraints. A weight reduction of 40% was obtained relative to the existing steel

structure. Additional savings could have been achieved by integrating composites into the initial system design instead of replacing an existing steel design.

The current steel foundation, shown in figure 13, is fabricated from welded high strength (HY-80) steel plate. A one-half scale composite model of the foundation was developed and analyzed using the finite element computer program ABAQUS. The idealized composite structure consists of 150 elements, 1160 nodes and 3500 degrees of freedom. The governing loads for foundation design and finite element analysis are derived from the shock requirements of MIL-S-901D. Stresses resulting from an equivalent static load of 40,000 pounds are presented in figure 14 to show the distribution of stresses in the composite model for the athwartship load case.

In selecting a vendor to fabricate the foundation model, cost was a major consideration. Rough cost estimates were obtained from a number of vendors representing different composite fabrication processes. The SCRIMP process (described earlier) was selected because cost estimates were 50% less than other methods. An epoxy resin was selected for the foundation based on strength and shrinkage considerations. The areas in the rounded corners of the foundation were up to 1.5 inches thick and were likely to delaminate if a low shrinkage resin was not used. An E-glass fabric (type 7781) was employed with an epoxy resin system. A quasi-isotropic ply stacking sequence was employed with predominately 0 degree and 0/90 degree layers near the mid thickness with increasing +/-45 degree layers near the surfaces. The added thickness in the corners was obtained by tapering in 15 plies of 0/90 degree cloth. The tapered areas on the foundation were created by dropping plies from the center most regions.

The basic SCRIMP fabrication process was described earlier. For the foundation, however, a plug having the geometry of the foundation model was fabricated from plywood and high density plastic foam. A glass-epoxy mold was then cast over the plug to form a female mold. Peel ply and fabric were laid into the mold, a vacuum was pulled on the fabric stack, and resin was injected into the layup by means of the SCRIMP process. The part was then cured at 225 degrees F for two hours. At this point the part was demolded and trimmed, and a post cure to 300 degrees F was performed. A photograph of the completed foundation is shown in figure 15.

The fabrication costs for the foundation model are much higher than would be incurred in an actual production situation on a dollars per pound basis. Since it was the first structural application of this resin system in the SCRIMP process, some development work was required. In addition, the full cost of the plug and mold were included in the foundation cost instead of being amortized over several parts as would be the case in production. The small size of the model actually increased the difficulty of the layup process, especially in placing the fabric in the corner areas. The layup time could have also been reduced by substantially decreasing the number of layers. In fact, it is felt that a high quality full-scale structure could be fabricated using one-half the number of plies that were employed for the foundation model. The cost of the foundation model was approximately \$90 per pound. It is estimated that full-scale foundations could be fabricated on a production basis for \$20 to \$40 per pound.

The strength of the composite foundation was evaluated by subjecting

it to high-impact shock tests (per MIL-S-901D) using a medium weight shock test machine. The foundation was tested in four different orientations: vertically, longitudinally, athwartships, and at an inclined angle of 30 degrees. The foundation, with a dummy steel mass to simulate the weight and center of gravity of supported equipment, is shown mounted in a vertical position on the medium weight shock machine in figure 16. Eight strain gages and from 2 to 5 accelerometers were recorded dynamically during each test. The foundation was tested both with and without resilient mounts which would be used between the foundation and the equipment bedplate on a full scale application. The composite foundation was able to sustain 16 impact blows, resulting in accelerations of up to 150 g's with no apparent degradation. In fact, eight additional tests were performed, going up to the maximum capacity of the test machine, to gain additional information on the integrity of the foundation. While a thorough examination has not yet been performed, there are no indications of structural degradation.

#### REPAIR USING 'SCRIMP' PROCESS

The above prototypes have clearly demonstrated the viability of the 'SCRIMP' process for the low-cost high-quality fabrication of these structures. It has also been recognized that it is important to demonstrate the effectiveness of the "SCRIMP" process when used as a repair procedure.

Such a capability has recently been successfully demonstrated on a sonar dome for the Hamilton Class of U.S. Coast Guard Cutters. These large GRP domes had been experiencing some delamination along the aft keel line upon internal pressurization with seawater. The Carderock Division, Naval Surface Warfare Center (CARDEROCKDIV) has been supporting the U.S. Coast Guard in investigating the cause of this delamination and in developing an improved structural dome configuration and repair procedure to correct the delamination problem. A structural modification to the dome has been developed which included increasing the radius of curvature at the bottom of the dome, increasing the laminate thickness from 0.75 inches to 1.0 inches, introducing + 45/-45 degree fabric plies into the 0/90 layup schedule, using vinyl ester resin instead of polyester to improve the toughness of the dome, and applying the 'SCRIMP' process. The repair procedure used on the dome is indicated in figure 17. The delaminated area was cutout, the laminate around the cutout was stepped back using a drop-off ratio of 20 to 1, and the stepped area was ground smooth. A male mold was then inserted into the cutout to provide a smooth contour for layup purposes. Dry glass fabric ( 9.6 oz.) was then 'scrimped' onto the dome in four steps, involving 28 plies per step. Four stages were required to minimize the development of wrinkles in the fabric under vacuum, and to allow them to be readily worked out when they did appear. After each step the applied laminate was allowed to cure at room temperature for 48 hours. After all 4 steps were complete, the repaired dome was allowed to post cure at 140 degrees F for 8 hours.

The repaired dome was subject to an internal pressurization to demonstrate that the repaired dome could withstand the required pressure of 17 psi without delamination. The dome actually withstood a pressure of 25 psi, approximately 50% higher than the design value

without any delamination.

The very successful repair demonstrated the capability for performing a high quality repair on a glass - polyester structure using a vinyl ester system. Although further tests are needed to validate the repair procedure for wider application, the results to date are very promising. The successful test also demonstrated that a thick laminate may be fabricated in stages using the 'SCRIMP' process. This has excellent implications for the multi- staged fabrication of thick (> 1 inch) composite structures for naval ship applications such as hulls.

#### POTENTIAL FUTURE APPLICATIONS

The excellent results to date with the 'SCRIMP' process for the deckhouse, mast, foundation and sonar dome structures, have stimulated interest in evaluating the process for application to the hulls of naval craft such as landing craft and patrol boats. Success could lead to the fabrication of hulls for smaller naval combatants, should they materialize in the future.

In an effort to evaluate the feasibility of using the 'SCRIMP' process for GRP hulls, CARDEROCKDIV intends to have a 0.35 scale validation model for a new landing craft concept, the Advanced Materiel Transporter (AMT), fabricated using this process. Under development for the U.S. Navy and Marine Corps, the full-scale AMT will be 126 feet long and 29 feet wide with a displacement of 249 long tons. The AMT will be capable of ballasting down and up to perform instream recovery of floating cargo and vehicles.

A hybrid structural design concept for the Validation Model and full scale AMT has been developed involving GRP single skin stiffened structure in the hull and balsa core sandwich structure in the deck and wing walls as shown in figure 18. The Validation Model will be constructed by fabricating the major structural elements, the hull, deck, transverse bulkheads and wing walls, using the 'SCRIMP' process and then assembling these elements using bolted and bonded joints. An effort will be made to lay up and scrimp the entire hull in a single step. If unsuccessful, the multi-step SCRIMP procedure used in the repair of the sonar dome will be applied. With the sonar dome procedure, dry GRP fabric will be laid down for one-half of the hull (transversely) and will be scrimped in place. After curing at room temperature for approximately 12 hours, the mold will be rotated and the process repeated. In this way, laminates of virtually any thickness may be built up. The GRP material will be overlapped along the keel for each stage of fabrication and should result in a primary - like bond based on the excellent results obtained in the sonar dome repair and test. In principle, it should be possible to use the 'SCRIMP' process to fabricate the 126 foot hull of the full scale AMT as a steppingstone to the hulls of larger naval craft in the future.

Some potentially near term applications for surface ships include composite maintenance enclosures, masts, and helicopter hangars. Near term submarine applications include the sail, control surfaces, sonar windows and stern cone. An agreement has recently been completed to fabricate a full scale model of a free flooding stern cone. This piece would be about 6 feet long and up to 1 inch thick and would be

fabricated in one piece (no joints). The proposed fabrication approach involves the placement of the fabric on a male mold, the insertion of the whole assembly into a female mold, the removal of the male mold, drawing the fiber pack against the female mold with a vacuum, and vacuum injection of the resin.

## SUMMARY AND CONCLUSIONS

Over the last 3 years, the U.S. Navy has been successful in evaluating and demonstrating a low cost vacuum RTM process, referred to as SCRIMP, for the fabrication of high performance composite ship structures of aerospace quality.

Some of the advantages of the SCRIMP process include:

- Low cost due to the elimination of pre-pregging, manual wet out, expensive tooling and autoclaves.
- High quality and superior mechanical properties due to high glass content (70%) and extremely low void content (< 1%).
- Mechanical properties that are competitive with properties achieved using wet lay-up or prepreg with autoclave cure.
- Versatility for fabricating large composite monocoque, single skin stiffened, and sandwich structures using a variety of fibers and resins. Vinyl ester, polyester, epoxy and phenolic have been demonstrated.
- Projected low costs in production of \$5 -\$8/lb for GRP deckhouse structure, \$20 to \$30/lb for composite masts of hybrid glass/ carbon construction and \$20 to \$40/lb for composite foundations using a system of E- glass and epoxy resin.

Using the SCRIMP process, the capability has been demonstrated for the design and fabrication of composite ship structures including deckhouses, masts and foundations. A comparison of estimated weight and production costs for these composite applications versus conventional metallic structures are summarized in Table 5. It is seen that the composite deckhouse and foundation structures should be able to be fabricated at less cost than the corresponding metallic structures. Some of the capabilities for design and fabrication are highlighted below:

- Fabrication of large tapered (8'x16') hat-stiffened and sandwich deckhouse panels of aerospace quality in one step.
- Fabrication of thick, geometrically complex, monocoque structure for masts and foundations.
- Design of affordable composite ship structures that meet air blast and shock requirements.
- Development of hybrid GRP/ steel structure to maximize structural performance and shipyard producibility.
- Repair of thick monocoque GRP structures.
- Multi- stage scrimping of thick laminates for potential application to the construction of large GRP hulls.

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Table 1 - Known Vacuum RTM Processes

| <u>COMPANY</u>             | <u>PROCESS</u>                       | <u>APPLICATION</u>                                                                |
|----------------------------|--------------------------------------|-----------------------------------------------------------------------------------|
| • SEEMANN COMPOSITES, INC. | SCRIMP                               | DECKHOUSE, MAST, FOUNDATION<br>SONAR DOME, RIB HULL STRUCTURE                     |
| • DOUGLAS AIRCRAFT CORP.   | VACUUM<br>IMPREGNATION               | THREE STRINGER WING PANELS                                                        |
| • JEREMY ROGERS, UK        | VARI                                 | 36' RACING BOATS (4)<br>34' PRODUCTION BOATS (~15)<br>40'+ PRODUCTION BOATS (~25) |
| • LE COMPTE HOLLAND BV     | VACUUM ASSISTED<br>INJECTION MOLDING | 22M LANDING CRAFT                                                                 |
| • AMI, INC.                | PRESTO-VAC                           | TRUCK AIR SHIELDS                                                                 |

Table 2 - Mechanical Property Comparison (Hand Lay-up vs. SCRIMP vs. Autoclave)

| PROPERTY                                   | HAND LAY-UP <sup>1</sup><br>50%*<br>POLY<br>WR | SCRIMP <sup>1</sup><br>70%*<br>POLY<br>WR | SCRIMP <sup>1</sup><br>70%*<br>VINYL<br>WR | SCRIMP <sup>2</sup><br>70%*<br>VINYL<br>CLOTH | PRE-PREG <sup>2</sup><br>AUTOCLAVE<br>EPOXY<br>CLOTH | SCRIMP <sup>3</sup><br>70%*<br>VINYL<br>3-TO-1 | WET LAY-UP<br>AUTOCLAVE<br>POLY<br>3-TO-1 |
|--------------------------------------------|------------------------------------------------|-------------------------------------------|--------------------------------------------|-----------------------------------------------|------------------------------------------------------|------------------------------------------------|-------------------------------------------|
| TENSILE<br>STRENGTH<br>(ksi)               | 46.8                                           | 67.3                                      | 70.9                                       | -                                             | -                                                    | 109.7                                          | 111.1                                     |
| TENSILE<br>MODULUS<br>(msi)                | 3.2                                            | 4.5                                       | 4.6                                        | -                                             | -                                                    | 5.5                                            | 5.5                                       |
| COMPRESSIVE<br>STRENGTH<br>(ksi)           | 39.1                                           | 42.1                                      | 68.6                                       | 61.0                                          | 60.0                                                 | 67.6                                           | 78.2                                      |
| FLEXURAL<br>STRENGTH<br>(ksi)              | 52.2                                           | 66.8                                      | 90.0                                       | 82.0                                          | 75.0                                                 | 121.4                                          | -                                         |
| INTERLAMINAR<br>SHEAR<br>STRENGTH<br>(ksi) | 3.3                                            | 4.5                                       | 7.3                                        | 7.2                                           | 8.5                                                  | 6.8                                            | 3.3                                       |

<sup>1</sup> 24 oz WOVEN ROVING  
<sup>2</sup> 8 HARNES SATIN CLOTH  
<sup>3</sup> TWILL WEAVE

\* APPROXIMATE BY WEIGHT

Table 3 - Comparison of Structural Performance for Deckhouse Modules

| MODULE        | MAX./ALLOWABLE DEFLECTION | MAX./ALLOWABLE STRAIN |
|---------------|---------------------------|-----------------------|
| GRP STIFFENED | 0.51 (*)                  | 0.50 (*)              |
|               | 1.12 (**)                 | 0.87 (**)             |
| GRP SANDWICH  | 0.96 (***)                | 0.96 (***)            |

(\*) MISERS GOLD EVENT (DESIGN PRESSURE)

(\*\*) DISTANT IMAGE EVENT (2 x DESIGN PRESSURE)

(\*\*\*) DISTANT IMAGE EVENT (DESIGN PRESSURE)

ALLOWABLE STRAIN = ULTIMATE STRAIN/1.25

ALLOWABLE DEFLECTION = 5.0 INCHES

Table 4 - Weight and Cost Comparison for Deckhouse Modules

| DECKHOUSE MODULE       | PANEL AREAL WEIGHT<br>(lb/ft <sup>2</sup> ) | MODULE WEIGHT<br>(lb) | PANEL COST |                       | MODULE COST<br>(\$K) |
|------------------------|---------------------------------------------|-----------------------|------------|-----------------------|----------------------|
|                        |                                             |                       | (\$/lb)    | (\$/ft <sup>2</sup> ) |                      |
| HAT STIFFENED          | 8.4                                         | 6500                  | 9.40       | 78.60                 | 87.5                 |
| FIRE-HARDENED SANDWICH | 6.5                                         | 5400                  | 15.90      | 103.50                | 89.1                 |



Table 5 - Weight and Cost Comparison for Metallic vs. Composite Ship Structures

| ITEM        | COMPARISON | WT SAVINGS<br>FOR<br>COMPOSITES | FABRICATION COST [\$ / LB]* |            | COMPONENT COST<br>[\$ COMPOSITE / \$ METAL]** |
|-------------|------------|---------------------------------|-----------------------------|------------|-----------------------------------------------|
|             |            |                                 | STEEL OR<br>ALUMINUM        | COMPOSITES |                                               |
| DECKHOUSES  | STEEL      | 35-45%                          | 4-6                         | 5-8        | 0.8                                           |
| MASTS       | ALUMINUM   | 20-50%                          | 9-12                        | 20-30      | 1.5                                           |
| FOUNDATIONS | STEEL      | UP TO 60%                       | 10-40                       | 20-40      | 0.5                                           |

\* ASSUMES COMPOSITES IN FULL PRODUCTION

\*\* BASED ON AVERAGE WEIGHT AND COST SAVINGS

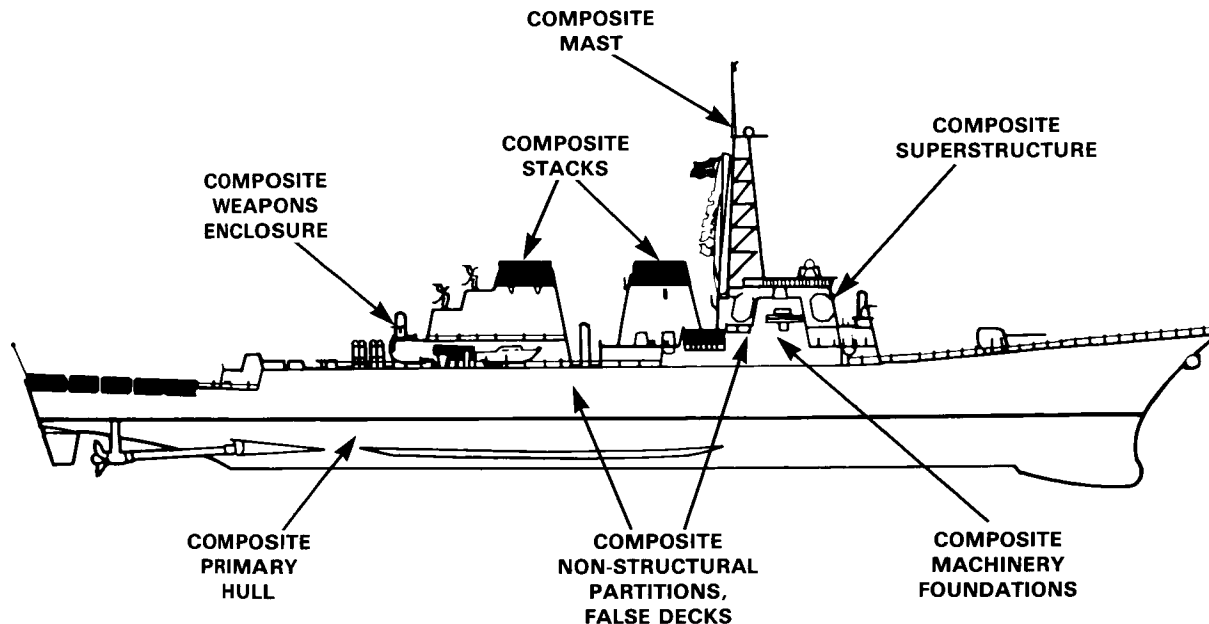


Figure 1 - Potential Surface Ship Applications of Composite Structures

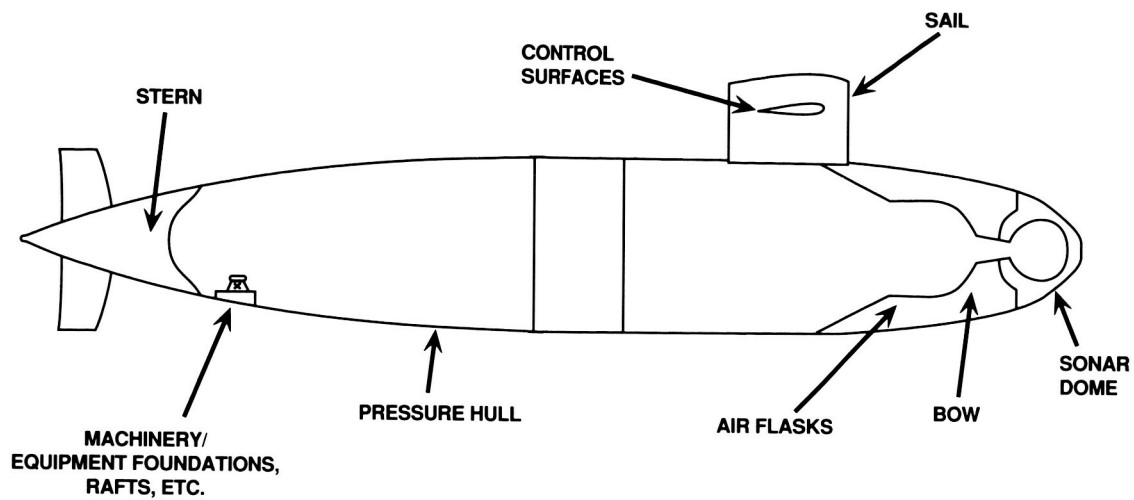


Figure 2 - Potential Submarine Applications of Composite Structures

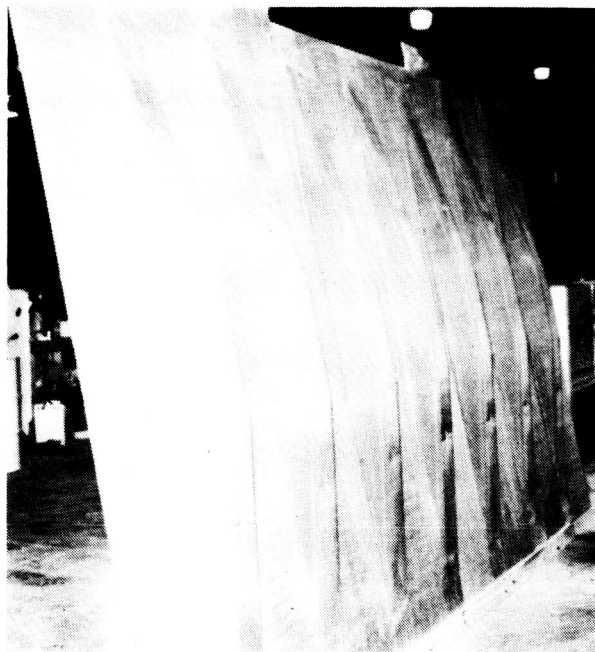
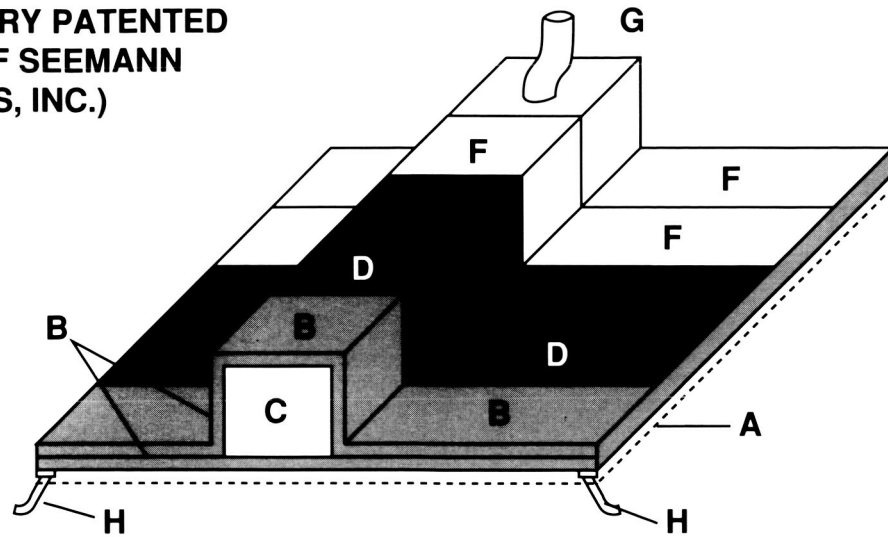


Figure 3. Photograph of GRP Stiffened Deckhouse Panel

(PROPRIETARY PATENTED  
PROCESS OF SEEMANN  
COMPOSITES, INC.)



|                 |                         |
|-----------------|-------------------------|
| A - TOOL        | E - DISTRIBUTION MEDIUM |
| B - DRY PREFORM | F - VACUUM BAG          |
| C - BALSA CORE  | G - RESIN INLET         |
| D - PEEL PLY    | H - VACUUM OUTLET       |

Figure 4 - SCRIMP Process for Single Skin Stiffened Panel

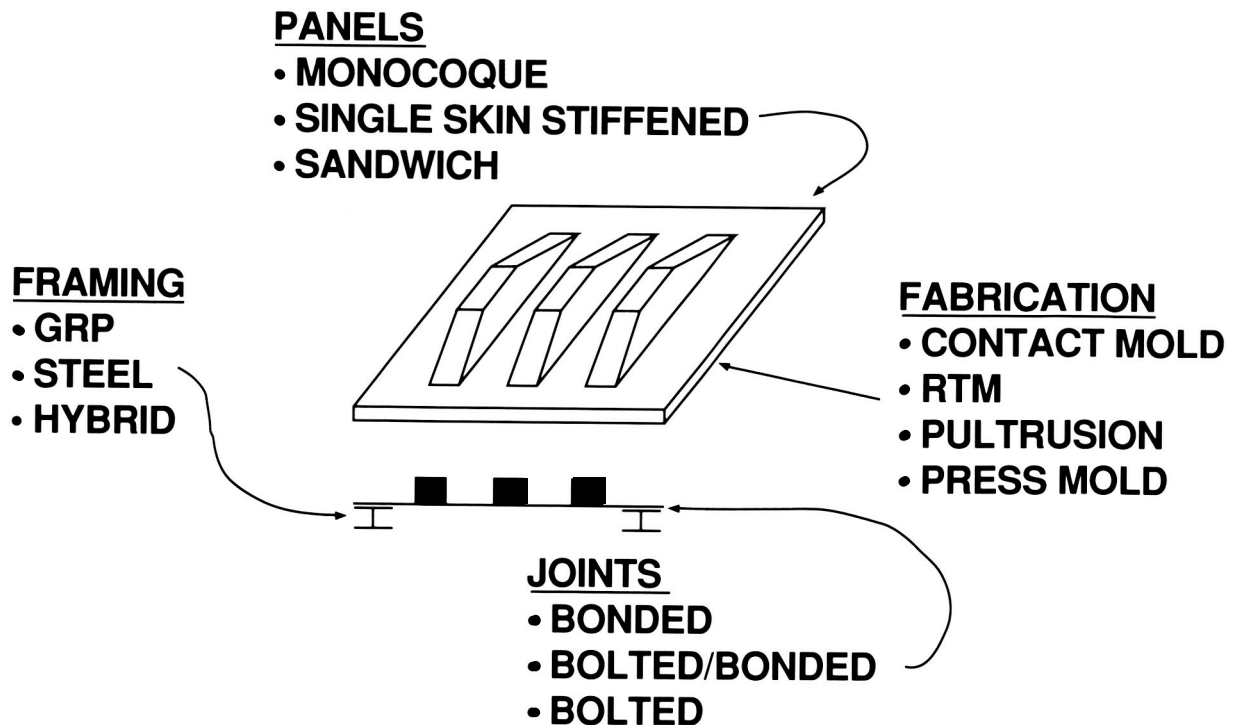


Figure 5 - Design/Fabrication Options for Composite Deckhouse Panels

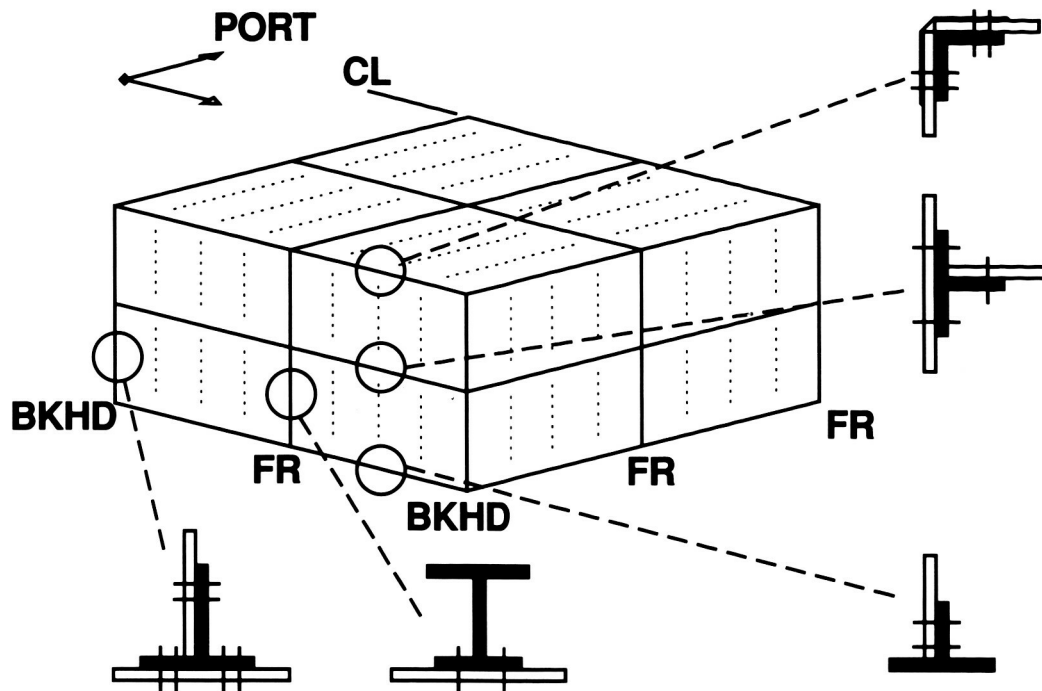
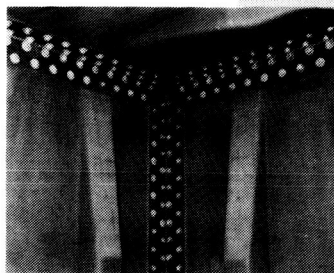
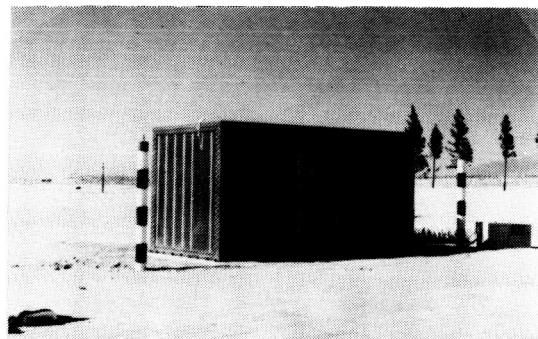
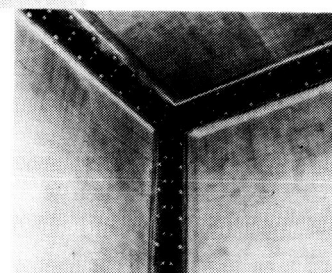


Figure 6 - Joint Details for GRP Deckhouse

(a) Exterior View of Deckhouse Module



(b) Inside Corner of Stiffened Module



(c) Inside Corner of Sandwich Modules

Figure 7. Photographs of GRP Stiffened and Sandwich Modules

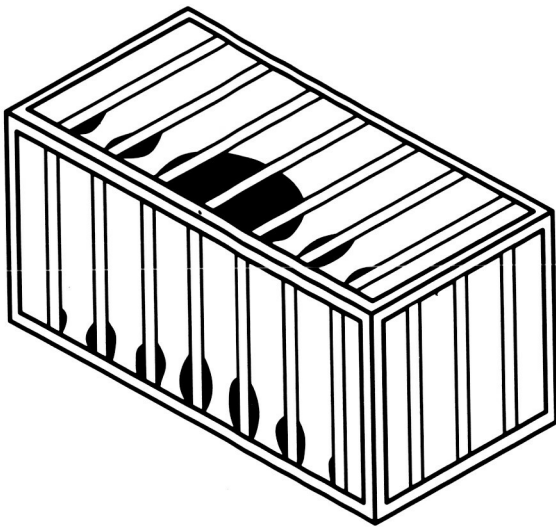


Figure 8. Delamination Map For GRP Stiffened Module

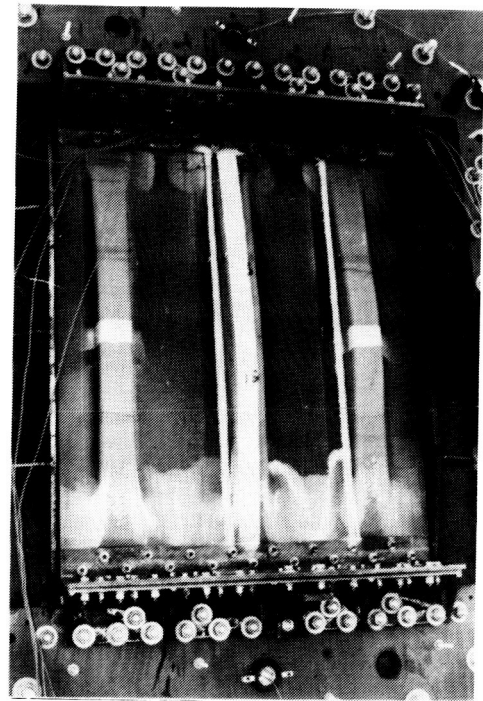


Figure 9. One-Half Scale Deckhouse Panel After Blast Tube Test

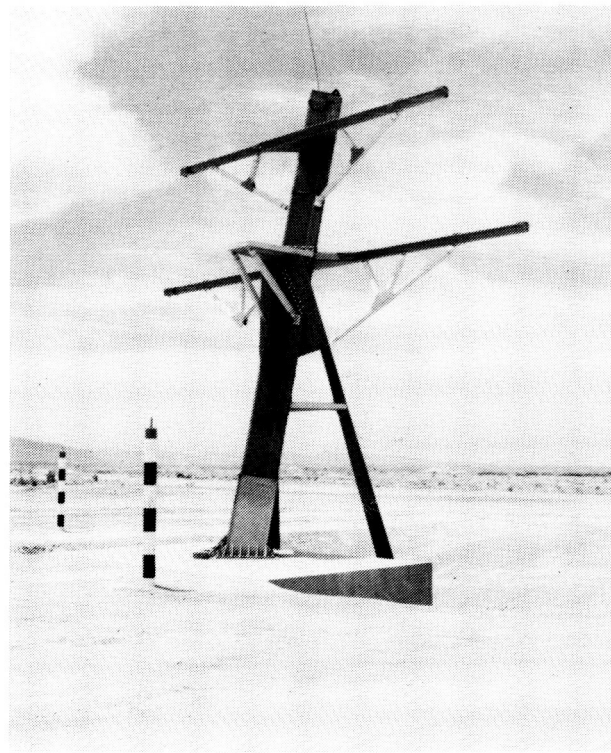


Figure 10. Photograph of One-Half Scale Composite Mast

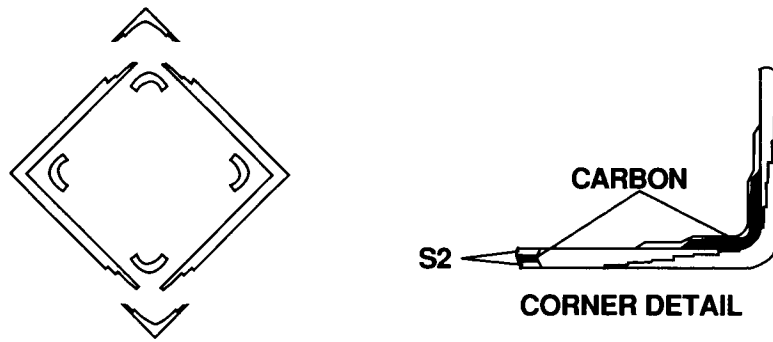


Figure 11 - Materials and Fabrication Details for Main Trunk



Figure 12 - Typical Cross-Section for Aft Legs and Yard Arms

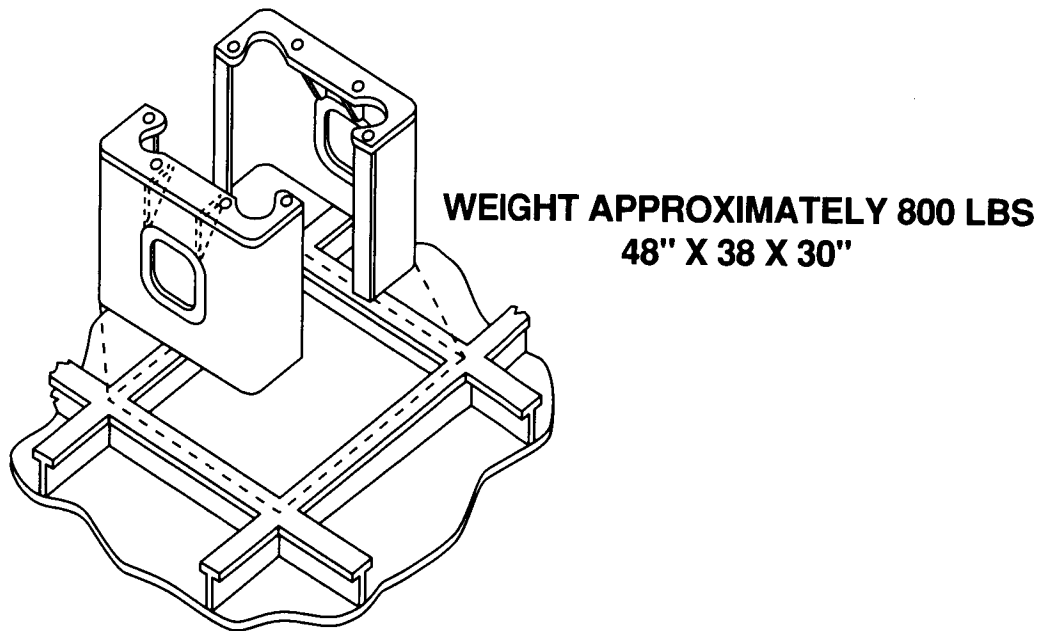


Figure 13 - Steel Fresh Water Pump Foundation

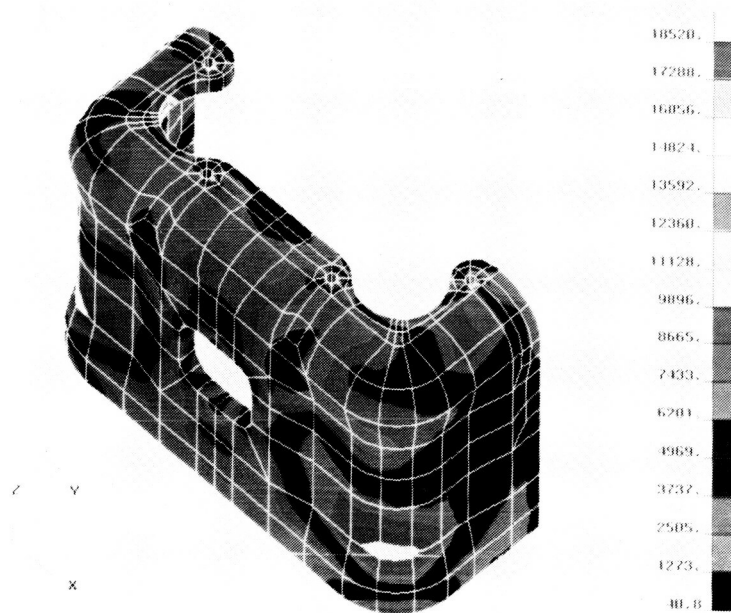


Figure 14. Stress Distribution in Composite Foundation

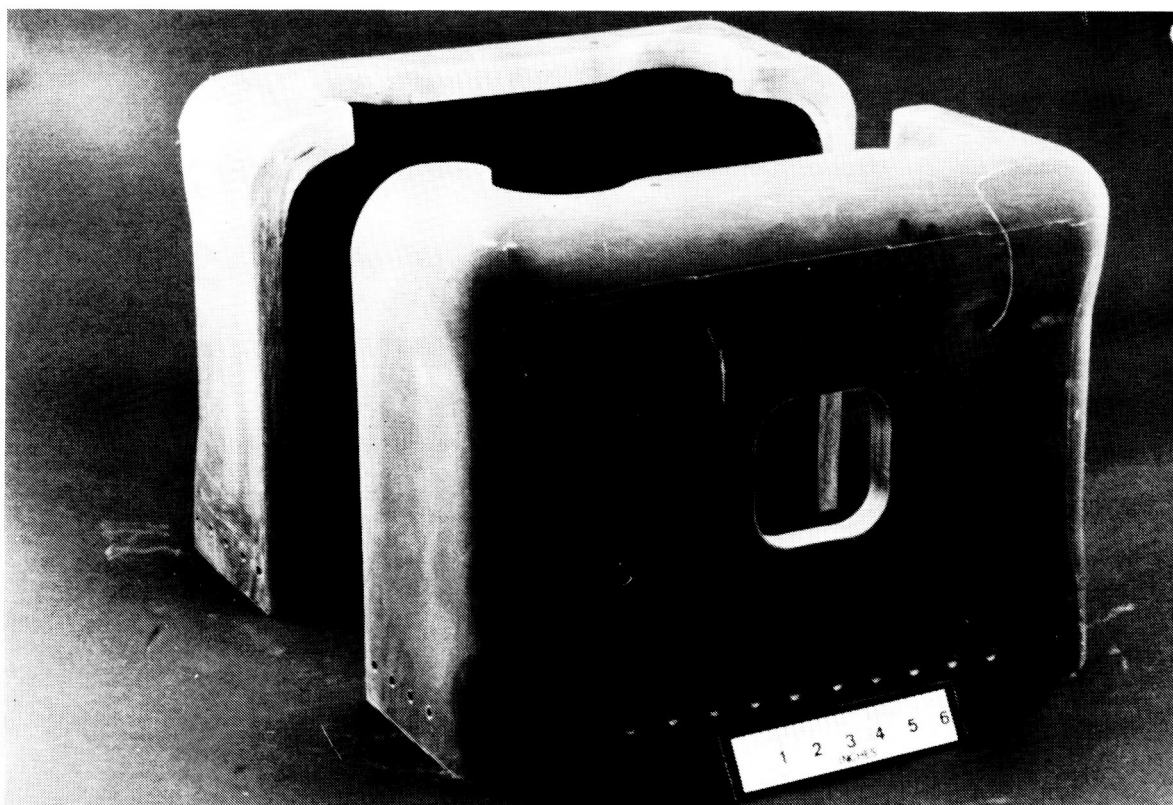


Figure 15. Photograph of Composite Submarine Pump Foundation (1/2 Scale)

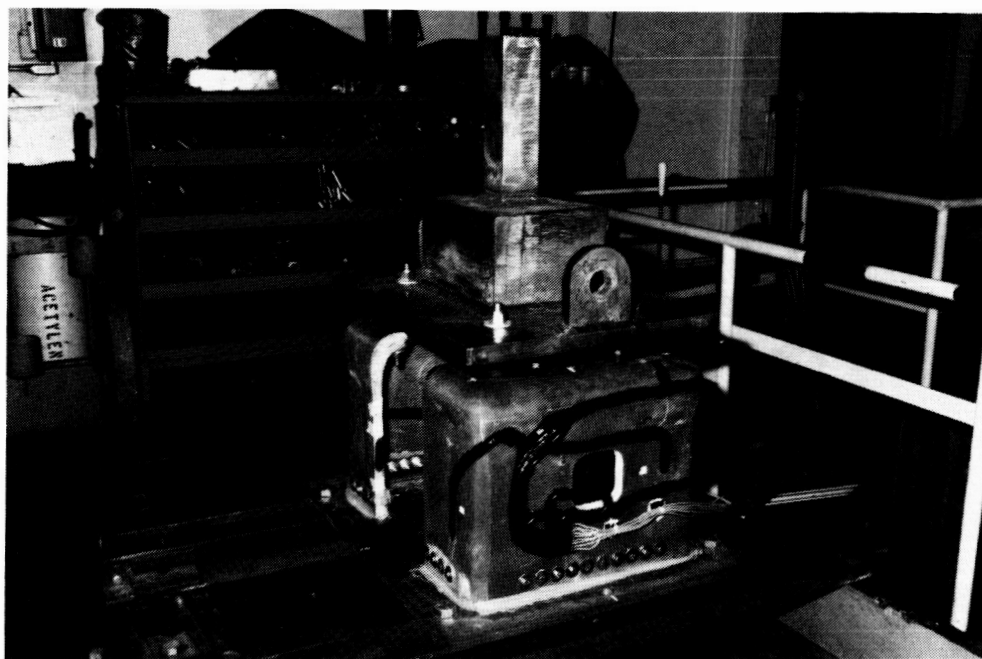


Figure 16. Composite Foundation on Medium Weight Shock Machine

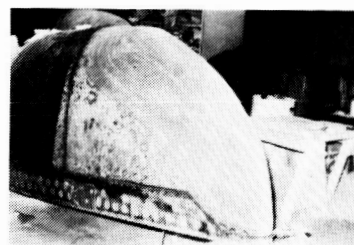
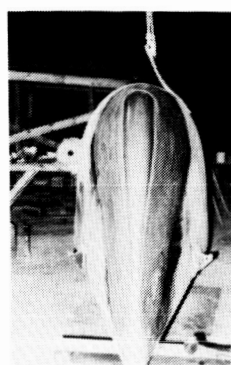
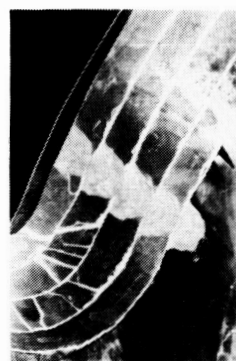
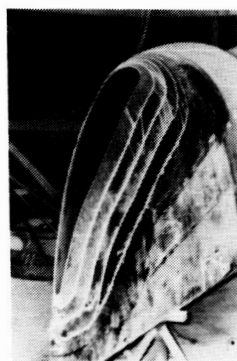


Figure 17. Repair of GRP Sonar Dome



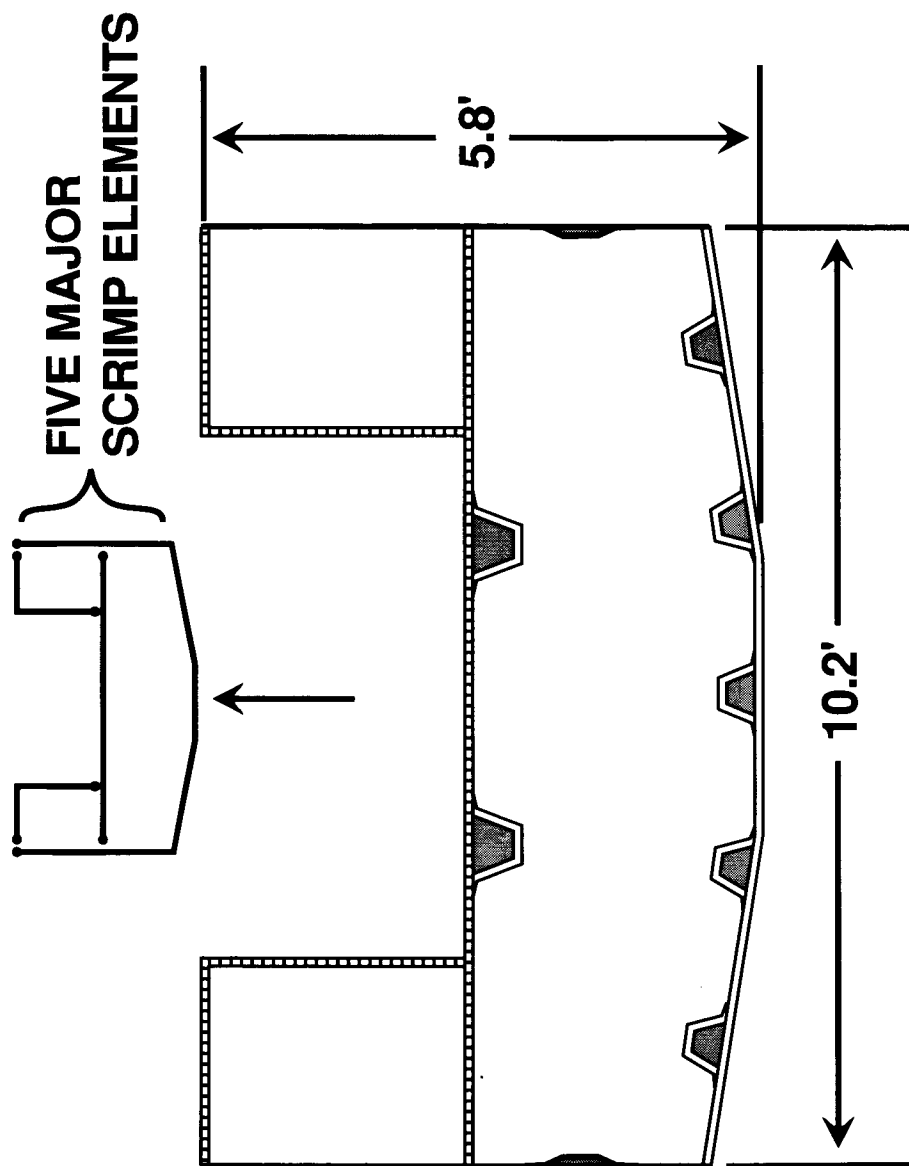


Figure 18 - Midship Design for Advanced Materiel Transporter

**SESSION X**

**AIRCRAFT DESIGN METHODOLOGY (B)**

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## Analysis of Stresses in Finite Anisotropic Panels with Centrally Located Cutouts

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SR0-39

### ABSTRACT

P- 21

A method for analyzing biaxial- and shear-loaded anisotropic rectangular panels with centrally located circular and elliptical cutouts is presented in the present paper. The method is based on Lekhnitskii's complex variable equations of plane elastostatics combined with a boundary collocation method and a Laurent series approximation. Results are presented for anisotropic panels with elliptical cutouts and subjected to combined shear and compression loading. The effects on the stress field of panel aspect ratio, anisotropy, cutout size, and cutout orientation are addressed. Angle-ply laminates, unidirectional off-axis laminates, and  $[(\pm 45/0/90)_3]_S$ ,  $[(\pm 45/0_2)_3]_S$ , and  $[(\pm 45/90_2)_3]_S$  laminates are examined.

### INTRODUCTION

Stress distributions in laminated composite panels with cutouts are an important consideration in aircraft design and analysis. Cutouts are often necessary in aircraft structures to form access ports for electrical and mechanical systems. In addition, significant weight savings can be achieved through the introduction of cutouts in wing ribs and other aircraft components. The effects of cutout shape and orientation on the magnitude and distribution of the stress field are important in the design of these components. Presently, the majority of stress analysis methods are based on classical infinite plate theory or finite element analysis. Finite element analysis has been a popular approach to the stress analysis of finite-dimensional panels with cutouts (refs. 1, 2). These analyses produce accurate results, but they are costly methods of performing parametric studies in which several different materials and geometries must be considered. The method described in the present paper is an alternative to finite element analysis for panels with cutouts.

The method presented herein is based on the complex variable equations of plane elastostatics presented by Lekhnitskii (ref. 3). These equations are used in conjunction with a boundary collocation method and a Laurent series approximation to analyze the stress fields of anisotropic panels with centrally located cutouts. This method allows for biaxial, shear, and combined loading, and accommodates elliptical cutouts of arbitrary orientation, as well as circular cutouts.

An analytical study of the stress distribution in anisotropic panels with cutouts is presented to illustrate the versatility of the method, and some comparisons are made with

experimental data. The effects of panel aspect ratio, anisotropy, cutout size, and cutout orientation on the stress field are examined. Angle-ply laminates, unidirectional off-axis laminates, as well as  $[(\pm 45/0/90)_3]_S$ ,  $[(\pm 45/0_2)_3]_S$ , and  $[(\pm 45/90_2)_3]_S$  laminates, are considered.

## SYMBOLS

|               |                                                                                                                                              |
|---------------|----------------------------------------------------------------------------------------------------------------------------------------------|
| $A$           | ellipse minor axis                                                                                                                           |
| $B_{kn}$      | constant coefficients of Laurent series                                                                                                      |
| $C_{mkn}$     | coefficients of Laurent series                                                                                                               |
| $D$           | constant coefficients                                                                                                                        |
| $E_x$         | circular cutout diameter and ellipse major axis                                                                                              |
| $E_y$         | average elastic modulus in x-direction                                                                                                       |
| $F$           | average elastic modulus in y-direction                                                                                                       |
| $F_m$         | Airy stress function                                                                                                                         |
| $G_{xy}$      | applied forces at panel boundaries                                                                                                           |
| $L$           | average shear modulus                                                                                                                        |
| $S$           | panel length                                                                                                                                 |
| $W$           | arc length on interior or exterior boundary                                                                                                  |
| $X_n$         | panel width                                                                                                                                  |
| $Y_n$         | x-component of boundary traction                                                                                                             |
| $z_k$         | y-component of boundary traction                                                                                                             |
| $\epsilon_x$  | complex variable defined as $z_k = x + \mu_k y$                                                                                              |
| $\eta_{xy,x}$ | average strain in x-direction                                                                                                                |
|               | coefficient of mutual influence of the second kind which characterizes shearing in the xy-plane caused by a normal stress in the x-direction |
| $\eta_{xy,y}$ | coefficient of mutual influence of the second kind which characterizes shearing in the xy-plane caused by a normal stress in the y-direction |
| $\theta$      | fiber orientation angle                                                                                                                      |
| $\mu_k$       | complex roots of the characteristic equation                                                                                                 |
| $\nu_{xy}$    | Poisson's ratio                                                                                                                              |
| $\Psi$        | inclination angle of elliptical cutout                                                                                                       |
| $\sigma_x$    | average normal stress in the x-direction                                                                                                     |
| $\sigma_y$    | average normal stress in the y-direction                                                                                                     |
| $\sigma_{xy}$ | average shear stress                                                                                                                         |
| $\phi_k$      | functions of $z_k$ which make up the stress function                                                                                         |
| $\Phi_k$      | first derivative of $\phi_k$                                                                                                                 |
| $\xi_i$       | transformation variable                                                                                                                      |

A bar over a quantity denotes its complex conjugate.

## ANALYSIS

### Theory

The objective of this analysis is to develop a method for determining the stress distribution in a finite anisotropic panel with a centrally located elliptical cutout. The panel is loaded by in-plane forces which do not vary through the thickness, and a state of generalized plane stress is assumed. Thus, average material properties are employed in the present stress analysis, which is based on Lekhnitskii's complex variable equations (ref. 3). A complete description of this analysis is provided in references 4 and 5, and a summary follows.

The generalized biharmonic equation for an anisotropic material in terms of an Airy stress function,  $F$ , and the average material properties is

$$\frac{\partial^4 F}{\partial y^4} - 2\eta_{xy,x} \frac{\partial^4 F}{\partial y^3 \partial x} - \left(2\nu_{xy} - \frac{E_x}{G_{xy}}\right) \frac{\partial^4 F}{\partial y^2 \partial x^2} - 2\eta_{xy,y} \frac{E_x}{E_y} \frac{\partial^4 F}{\partial y \partial x^3} + \frac{E_x}{E_y} \frac{\partial^4 F}{\partial x^4} = 0$$

This biharmonic equation can be simplified using the transformation

$$z_1 = x + \mu_1 y \quad z_2 = x + \mu_2 y$$

where  $\mu_1$ ,  $\mu_2$ , and their complex conjugates are the roots of the characteristic equation

$$\mu^4 - 2\eta_{xy,x} \mu^3 + \left(\frac{E_x}{G_{xy}} - 2\nu_{xy}\right) \mu^2 - 2\eta_{xy,y} \frac{E_x}{E_y} \mu + \frac{E_x}{E_y} = 0$$

Using the above transformation, the generalized biharmonic equation can be written as

$$\frac{\partial}{\partial z_1} \frac{\partial}{\partial z_2} \frac{\partial}{\partial \bar{z}_1} \frac{\partial}{\partial \bar{z}_2} F = 0$$

The solution for the stress function,  $F$ , is

$$F = \phi_1(z_1) + \phi_2(z_2) + \overline{\phi_1(z_1)} + \overline{\phi_2(z_2)}$$

Applied displacements or tractions along the panel boundary can be related to the complex-valued stress function. In this paper, only traction boundary conditions are considered. The boundary tractions in the  $x$ -direction,  $X_n$ , and in the  $y$ -direction,  $Y_n$ , are related to the stress function by

$$2\operatorname{Re} [\Phi_1(z_1) + \Phi_2(z_2)] \Big|_{z_0}^z = \pm \left( - \int_0^S Y_n dS \right)$$

$$2\operatorname{Re} [\mu_1 \Phi_1(z_1) + \mu_2 \Phi_2(z_2)] \Big|_{z_0}^z = \pm \int_0^S X_n dS$$

where  $S$  is the length of the boundary arc that begins at a point  $z_0$  and ends at  $z$ , and  $\Phi_k(z_k)$  is defined as

$$\Phi_k(z_k) = \frac{\partial \phi_k}{\partial z_k}$$

If the value of this stress function is known for every point within the panel boundaries, the stress distribution in the panel can be determined. In the present analysis, this stress function is represented by a truncated Laurent series containing unknown constant coefficients:

$$\Phi_k(z_k) = \sum_{N}^{-N} B_{kn} z_k^n$$

The resultant force on every arc of the interior and exterior boundaries is known and, therefore, Lekhnitskii's force equations can be used to solve for the unknown coefficients through boundary collocation. Satisfying the force loading conditions along the interior and exterior panel boundaries results in a system of equations that can be solved for the unknown Laurent series constants:

$$[C_{mkn}] \{B_{kn}\} = \{F_m\}$$

where  $B_{kn}$  are the unknown Laurent series constants,  $C_{mkn}$  are the coefficients of the Laurent series constants, and  $F_m$  are the applied force resultants. In order to improve solution convergence without increasing computation time, a least squares approach is implemented. The least squares method allows for an increase in the number of force equations

considered without an increase in the number of terms in the Laurent series. In the present case, twice as many equations as unknowns are considered; therefore,

$$\begin{aligned} [C_{mkn}] &\text{ is a } 16N \times 8N \text{ matrix} \\ \{B_{kn}\} &\text{ is an } 8N \text{ vector} \\ \{F_m\} &\text{ is a } 16N \text{ vector} \end{aligned}$$

To solve the system of equations for the unknown Laurent series coefficients, each side of the matrix equation is multiplied by the transpose of  $[C_{mkn}]$ . To improve the conditioning of the system matrix, the following mapping function which maps all of the points inside the panel boundaries to the exterior of a unit circle is used:

$$\xi_i = \frac{1}{A - i\mu_i D} \left( z_i + \sqrt{z_i^2 - A^2 - \mu_i^2 D^2} \right) \quad i = 1, 2$$

where  $D$  is the major axis of the elliptical cutout, and  $A$  is the minor axis. The use of this mapping function eliminates the problem of small numbers being raised to high powers. After the system of linear algebraic equations is solved for the Laurent series constants, the average stresses in the panel can be calculated using the following stress equations:

$$\sigma_x = \frac{\partial^2 F}{\partial y^2} = 2\text{Re} [\mu_1^2 \Phi_1'(z_1) + \mu_2^2 \Phi_2'(z_2)]$$

$$\sigma_y = \frac{\partial^2 F}{\partial x^2} = 2\text{Re} [\Phi_1'(z_1) + \Phi_2'(z_2)]$$

$$\sigma_{xy} = -\frac{\partial^2 F}{\partial x \partial y} = -2\text{Re} [\mu_1 \Phi_1'(z_1) + \mu_2 \Phi_2'(z_2)]$$

### Model

A FORTRAN program was written to implement the present analysis. The configuration analyzed consists of a rectangular panel with a centrally located cutout as shown in figure 1. The panel is of length  $L$  (in the  $x$ -direction) and of width  $W$ . The panel has either a circular or an elliptical cutout at its center. The circular cutout is of diameter  $D$ ; the elliptical cutout has a major axis  $D$  and a minor axis  $A$ . The major axis of the ellipse may be



inclined at some angle  $\Psi$  to the y-axis or may be aligned with the y-axis. The panel is subjected to an applied shear stress,  $\sigma_{xy}^0$ , and/or an applied compressive stress,  $\sigma_x^0$ . In this study, when both shear and compression stresses are applied, they are equal in magnitude. The boundary of the cutout is assumed to be traction-free. The lamina fiber angles,  $\theta$ , are measured with respect to the x-axis.

## RESULTS AND DISCUSSION

### Comparison with Experiment

To assess the accuracy of the current analytical approach, comparisons are made between experimental data and analytical results. The stress distribution near a circular cutout in a rectangular orthotropic panel with a uniaxial applied compressive stress is shown in figure 2. The  $(\pm 45/0_2/\pm 45/0_2/\pm 45/0/90)_{2S}$  graphite-epoxy laminate has a width of 4.49 inches, a length of 10 inches, and cutout-diameter-to-panel-width ratios,  $D/W$ , of 0.11, 0.17, 0.22, and 0.33. Normal stress along the y-axis,  $\sigma_x(0,y)$ , normalized by the applied compressive stress,  $\sigma_x^0$ , is plotted as a function of the distance from the center of the panel,  $y$ , normalized by half the panel width,  $W/2$ . The analytical results are represented by solid lines, and the experimental data are represented by symbols. The experimental data were obtained from a study presented in reference 6. The analytical results agree well with the experimental data.

A comparison between analytical and experimental data is also made for a square  $[(\pm 45/0/90)_3]_S$  graphite-thermoplastic laminate subjected to an applied shear stress,  $\sigma_{xy}^0$ . The panel has a width of 12 inches and cutout-diameter-to-panel-width ratios,  $D/W$ , of 0.063, 0.125, and 0.25. The normal strain along the y'-axis,  $\epsilon_x(0,y')$ , is plotted in figure 3 as a function of the nondimensional distance from the center of the panel,  $y'/W$ . For the purpose of this comparison, the panel is oriented at a  $45^\circ$  angle to the y-axis as shown in figure 3. The analytical results are represented by solid lines, and the experimental data from reference 7 are represented by symbols. The analytical results agree reasonably well with the experimental data; differences between analytical and experimental results are suspected to be largely due to the manner in which the load is introduced. In the analysis, a uniform shear stress is applied to the panel boundaries. In the experiment, loads are introduced through the use of a picture frame test fixture, which tends to concentrate the loads at the panel corners.

### Analytical Study

To demonstrate the capabilities of the analytical method, stresses are calculated for a range of panel dimensions, ply layups, and loading conditions. Results are presented in terms of a maximum stress value which is normalized by the corresponding applied stress. This maximum stress value is obtained by conducting a survey of stress values at numerous evenly-spaced points on the panel. Points inside the cutout region are excluded. The

laminates analyzed are made of Hercules, Inc. AS4/3502 graphite-epoxy unidirectional tape and the lamina properties for the laminates studied are presented in Table 1.

The FORTRAN program which implements the present method consists of two lines of input describing the panel geometry, loading conditions, and average material properties. Due to the minimal amount of input, the following cases require very little user time for the creation of the models. In addition, the program run time is short enough that the program can be run interactively. Consequently, the present method exhibits a considerable advantage in time savings over finite element analyses in a study of this nature.

$[(\pm 45)_6]_S$  and  $(45)_{24}$  laminates with circular cutouts. - As an example of the capabilities of the current analytical method, the extreme case of a unidirectional  $(45)_{24}$  laminate exhibiting shear-extension coupling is analyzed and the results are compared with those for a more practical  $[(\pm 45)_6]_S$  angle-ply laminate. The laminates contain circular cutouts and are subjected to a uniaxial compression load. The maximum normal stress,  $(\sigma_x)_{\text{maximum}}$ , normalized by the applied compressive stress,  $\sigma_x^0$ , is plotted in figure 4 as a function of panel aspect ratio,  $L/W$ , for different cutout diameters. The ratio  $L/W$  varies from one to three, and  $D/W$  is equal to 0.1, 0.3, and 0.6. The results for the  $[(\pm 45)_6]_S$  laminate are represented by the solid line, the results for the  $(45)_{24}$  laminate are represented by the dashed line, and the symbols, shown at points where analytical values are calculated, represent the different cutout diameters. As the cutout size increases, the panel aspect ratio has a significant effect on the maximum normal stresses, which are much higher for low aspect ratio panels than for high aspect ratio panels. The maximum normal stresses increase for large cutout sizes, and maximum normal stresses in the  $(45)_{24}$  laminates are greater than maximum normal stresses in the  $[(\pm 45)_6]_S$  laminates.

The analytical results for the  $[(\pm 45)_6]_S$  and  $(45)_{24}$  laminates subjected to an applied shear stress are shown in figure 5. The maximum shear stress,  $(\sigma_{xy})_{\text{maximum}}$ , normalized by the applied shear stress,  $\sigma_{xy}^0$ , is plotted as a function of panel aspect ratio,  $L/W$ , for  $D/W$  equal to 0.1, 0.3, and 0.6. Panel aspect ratio does not significantly influence the magnitude of the maximum shear stresses in a panel with a small cutout but becomes increasingly important as the cutout diameter grows larger. As with the compression-loaded laminates, the  $(45)_{24}$  laminate has higher maximum shear stresses than the  $[(\pm 45)_6]_S$  laminate under shear loading for the range of panel aspect ratios investigated in the present study.

Analytical results are also calculated for the  $[(\pm 45)_6]_S$  and  $(45)_{24}$  laminates subjected to combined compression and shear. The maximum normal stress,  $(\sigma_x)_{\text{maximum}}$ , normalized by the applied compressive stress,  $\sigma_x^0$ , is plotted in figure 6; and the maximum shear stress,  $(\sigma_{xy})_{\text{maximum}}$ , normalized by the applied shear stress,  $\sigma_{xy}^0$ , is plotted in figure 7. The maximum shear and normal stresses are plotted as functions of panel aspect ratio. An examination of the results for the laminates loaded in shear only and the laminates subjected to combined shear and compression loading reveals that the addition of the compression load to the shear-loaded panels does not greatly impact the trends and magnitudes of the maximum shear stresses. However, for combined loading conditions, the maximum normal stresses exhibit behavior radically different from the maximum normal stresses in the laminates loaded in compression only (see fig. 4). Under combined shear and compression loading, the maximum normal stresses in the  $(45)_{24}$  laminates with  $D/W$  equal to 0.1 are lower than the maximum normal stresses in the  $[(\pm 45)_6]_S$  laminates with the same  $D/W$ . For large cutouts and low panel aspect ratios, the maximum normal stresses in the  $(45)_{24}$  laminates are larger

than the maximum normal stresses in the  $[(\pm 45)_6]_S$  laminates. However, for large cutouts and high aspect ratios, the maximum normal stresses in the  $[(\pm 45)_6]_S$  laminates are higher than the maximum normal stresses in the  $(45)_{24}$  laminates.

$[(\pm 45/0/90)_3]_S$ ,  $[(\pm 45/0_2)_3]_S$  and  $[(\pm 45/90_2)_3]_S$  laminates with circular cutouts.- In order to study the effects of panel aspect ratio and cutout size on the stress field of some commonly used laminates,  $[(\pm 45/0/90)_3]_S$ ,  $[(\pm 45/0_2)_3]_S$ , and  $[(\pm 45/90_2)_3]_S$  laminates subjected to compression, shear, and combined loads are analyzed. The maximum normal stress,  $(\sigma_x)_{\text{maximum}}$ , normalized by the applied compressive stress,  $\sigma_x^0$ , is plotted in figure 8 as a function of panel aspect ratio for these three laminates subjected to a compression load for  $D/W$  equal to 0.1, 0.3, and 0.6. The  $0^\circ$  plies have the highest extensional modulus and, therefore, the  $[(\pm 45/0_2)_3]_S$  laminate, which contains the most  $0^\circ$  plies of the laminates studied, has the highest maximum normal stresses for all cutout diameters. The  $[(\pm 45/90_2)_3]_S$  laminate, which does not contain any  $0^\circ$  plies, has the lowest maximum normal stresses for all values of  $D/W$ . The panel aspect ratio,  $L/W$ , has a minimal effect on the results for laminates with  $D/W$  equal to 0.1. However, as the cutout diameter increases, the laminates with larger cutout diameters have very high maximum normal stresses at low panel aspect ratios, and these maximum normal stress values tend to coalesce as the panel aspect ratio increases.

The maximum shear stress,  $(\sigma_{xy})_{\text{maximum}}$ , normalized by the applied shear stress,  $\sigma_{xy}^0$ , is plotted in figure 9 as a function of panel aspect ratio for the  $[(\pm 45/0/90)_3]_S$ ,  $[(\pm 45/0_2)_3]_S$ , and  $[(\pm 45/90_2)_3]_S$  laminates subjected to a shear load. For cutout-diameter-to-panel-width ratios,  $D/W$ , of 0.1 and 0.3, the  $[(\pm 45/90_2)_3]_S$  laminate has the highest maximum shear stresses and the  $[(\pm 45/0/90)_3]_S$  laminate has the lowest maximum shear stresses for the range of  $L/W$  considered. As the panel aspect ratio increases, this trend remains unchanged for  $D/W$  equal to 0.1 and 0.3. However, for  $D/W$  equal to 0.6, the ordering of the  $[(\pm 45/0/90)_3]_S$ ,  $[(\pm 45/0_2)_3]_S$ , and  $[(\pm 45/90_2)_3]_S$  laminates in terms of the highest and lowest maximum shear stresses changes as the panel aspect ratio increases. For low values of  $L/W$ , the same trend that is observed for laminates with  $D/W$  equal to 0.1 and 0.3 occurs for  $D/W$  equal to 0.6. As the panel aspect ratio begins to increase, the  $[(\pm 45/90_2)_3]_S$  laminate continues to have the highest maximum shear stresses, but the  $[(\pm 45/0_2)_3]_S$  laminate has the lowest maximum shear stresses. For a panel aspect ratio of three, the  $[(\pm 45/0_2)_3]_S$  laminate has the highest maximum shear stress, and the  $[(\pm 45/0/90)_3]_S$  laminate has the lowest maximum shear stress.

Results are also obtained for the  $[(\pm 45/0/90)_3]_S$ ,  $[(\pm 45/0_2)_3]_S$ , and  $[(\pm 45/90_2)_3]_S$  laminates subjected to combined shear and compression loads. The maximum normal stress,  $(\sigma_x)_{\text{maximum}}$ , normalized by the applied compressive stress,  $\sigma_x^0$ , is plotted as a function of panel aspect ratio in figure 10. The addition of the shear load to the compression load increases the maximum normal stresses. For all values of  $D/W$ , the  $[(\pm 45/0_2)_3]_S$  laminate still has the highest maximum normal stresses, and the  $[(\pm 45/90_2)_3]_S$  laminate still has the lowest maximum normal stresses. However, the change in the magnitude of the maximum normal stresses as the panel aspect ratio increases is not as large for the combined load case as for the compression load case. Maximum shear stresses,  $(\sigma_{xy})_{\text{maximum}}$ , normalized by the applied shear stress,  $\sigma_{xy}^0$ , are plotted as a function of panel aspect ratio in figure 11. The addition of the compression load to the shear load increases the maximum shear stresses in the laminates, but the general trends remain the same with one exception. For panel aspect ratios near 1.0, the  $[(\pm 45/0_2)_3]_S$  laminate has higher maximum shear stresses than the  $[(\pm 45/90_2)_3]_S$  laminate for  $D/W$  equal to 0.3 and 0.6. As expected from symmetry

considerations, the  $[(\pm 45/90)_2]_S$  and  $[(\pm 45/0)_2]_S$  laminates have identical maximum shear stress values for a panel aspect ratio of 1.0 when they are subjected to shear loading only (see fig. 9).

**Angle-ply laminates with circular cutouts.**- Angle-ply laminates with circular cutouts are analyzed in order to examine the effects of fiber angle orientation on the maximum stresses in panels subjected to shear, compression, and combined loads. In figure 12, maximum normal stresses,  $(\sigma_x)_{\text{maximum}}$ , normalized by the applied compressive stress,  $\sigma_x^0$ , are plotted as a function of fiber angle,  $\theta$ , for square, compression-loaded, angle-ply laminates with circular cutouts. The solid lines represent analytical results for  $D/W$  equal to 0.1, 0.3, and 0.6. The fiber angle,  $\theta$ , for the  $[(\pm\theta)_6]_S$  angle-ply laminate ranges from  $0^\circ$  to  $90^\circ$ . The maximum normal stresses are largest for a fiber angle of  $0^\circ$ , which coincides with the direction of the applied stress. The maximum normal stresses decrease steadily as the fiber angle is increased. This reduction in maximum normal stress values is attributed to the decrease in the extensional modulus of the angle-ply laminate as the fiber angle increases.

The maximum shear stresses,  $(\sigma_{xy})_{\text{maximum}}$ , normalized by the applied shear stress,  $\sigma_{xy}^0$ , are plotted in figure 13 as a function of fiber angle,  $\theta$ , for square, shear-loaded, angle-ply laminates with circular cutouts. The ratio  $D/W$  is equal to 0.1, 0.3, and 0.6, and  $\theta$  ranges from  $0^\circ$  to  $90^\circ$ . In some cases, the exact maximum stress is probably not found because the laminate stresses are calculated at a finite number of points on the panel. This discretization problem can cause some waviness in the maximum stress curves. The largest maximum shear stresses occur for a fiber angle of about  $45^\circ$  and decrease to minimums at fiber angles of  $0^\circ$  and  $90^\circ$ . This behavior appears to be consistent with the decrease in the laminate shear modulus as the fiber angle changes from  $45^\circ$ .

Square angle-ply laminates with circular cutouts with  $D/W$  equal to 0.1, 0.3, and 0.6 are also analyzed for combined shear and compression loads. The maximum normal stress,  $(\sigma_x)_{\text{maximum}}$ , normalized by the applied compressive stress,  $\sigma_x^0$ , and the maximum shear stress,  $(\sigma_{xy})_{\text{maximum}}$ , normalized by the applied shear stress,  $\sigma_{xy}^0$ , are plotted as functions of fiber angle in figures 14 and 15, respectively. The maximum normal and shear stresses show significant increases in magnitude over the maximum normal and shear stresses for the single load cases. The highest maximum normal stresses occur at a fiber angle of  $0^\circ$  as they did in the laminates subjected to a compression load only. The highest maximum shear stress for laminates with cutout-diameter-to-panel-width ratios,  $D/W$ , equal to 0.1 occurs at a  $45^\circ$  fiber angle and shifts toward a fiber angle of  $40^\circ$  as  $D/W$  increases to 0.6.

**Angle-ply laminates with elliptical cutouts.**- Angle-ply laminates with elliptical cutouts are examined to assess the effects of cutout shape, size, and orientation on the maximum stresses. For this analysis, the ratio of the major axis of the ellipse,  $D$ , to the panel width,  $W$ , is given values of 0.1, 0.3, and 0.6. The ratio of the minor axis,  $A$ , to the major axis of the ellipse,  $D$ , is equal to 0.5, 0.75, and 1.0. The major axis of the elliptical cutout is inclined at an angle  $\Psi$  to the  $y$ -axis, and  $\Psi$  ranges from  $0^\circ$  to  $90^\circ$ . The maximum normal stress,  $(\sigma_x)_{\text{maximum}}$ , normalized by the applied compressive stress,  $\sigma_x^0$ , is plotted as a function of  $\Psi$  for square, compression-loaded  $[(\pm 30)_6]_S$ ,  $[(\pm 45)_6]_S$ , and  $[(\pm 60)_6]_S$  laminates in figures 16, 17, and 18, respectively. Similar to the results for the angle-ply laminates with circular cutouts, the maximum normal stresses are highest in the  $[(\pm 30)_6]_S$  laminate and decrease as the fiber angle increases. Laminates having elliptical cutouts inclined at small angles to the  $y$ -axis

have the largest maximum normal stresses, and, in most cases, these maximum normal stresses increase as  $A/D$  decreases. As  $\Psi$  is increased, the maximum normal stresses decrease in all of the laminates with elliptical cutouts. At large values of  $\Psi$  the maximum normal stresses decrease as  $A/D$  decreases.

Maximum shear stresses,  $(\sigma_{xy})_{\text{maximum}}$ , for the  $[(\pm 30)_6]_S$ ,  $[(\pm 45)_6]_S$ , and  $[(\pm 60)_6]_S$  laminates subjected to shear loads are normalized by the applied shear stress,  $\sigma_x^0$ , and plotted as a function of the elliptical cutout orientation angle,  $\Psi$ , in figures 19, 20, and 21, respectively. The normalized maximum shear stress curves behave in a somewhat erratic manner. As mentioned previously, the laminate stresses are surveyed at a finite number of points. Therefore, the exact maximum stress is probably not always found, and some waviness in the maximum stress curves results. This waviness is magnified by the sensitivity of the value and location of the maximum shear stress to changes in  $\Psi$ , especially at low values of  $A/D$ . An example of this phenomenon is shown for the  $[(\pm 45)_6]_S$  laminate in figures 22 through 25. The shear stress distribution and the maximum shear stress location for laminates with the major axis of the elliptical cutout inclined at  $0^\circ$ ,  $25^\circ$ , and  $45^\circ$  are shown in figures 22, 23, and 24, respectively. In contrast, for circular cutouts the maximum shear stress location remains fixed. For example, the shear stress distribution for the  $[(\pm 45)_6]_S$  laminate with a circular cutout is shown in figure 25.

Although the maximum shear stress curves do not vary smoothly, the general trends are discernable. In the case of the  $[(\pm 30)_6]_S$  laminate, the maximum shear stresses occur when  $\Psi$  is about  $30^\circ$  for the smaller cutouts and about  $45^\circ$  for the largest cutout size. In the  $[(\pm 45)_6]_S$  laminate, the maximum shear stresses occur when  $\Psi$  is equal to  $45^\circ$  for all cutout sizes. Under shear loading conditions, the maximum shear stresses in the  $[(\pm 60)_6]_S$  laminate occur when  $\Psi$  is about  $60^\circ$  for small cutouts and about  $45^\circ$  for the largest cutout size. Of the three angle-ply laminates examined, the  $[(\pm 45)_6]_S$  laminate has the overall highest maximum shear stresses. In all of the angle-ply laminates with cutout-major-axis-to-panel-width ratios,  $D/W$ , equal to 0.1 and 0.3, the maximum shear stresses increase as  $A/D$  decreases. For the laminates with  $D/W$  equal to 0.6, the maximum shear stress decreases as  $A/D$  decreases for values of  $\Psi$  near  $0^\circ$  and  $90^\circ$ . For some range of values of  $\Psi$  between  $5^\circ$  and  $85^\circ$ , depending on the particular laminate, the maximum shear stresses are highest for  $A/D$  equal to 0.5, and lowest for  $A/D$  equal to 0.75, with the maximum shear stresses for  $A/D$  equal to 1.0 falling between the two.

The maximum normal stresses,  $(\sigma_x)_{\text{maximum}}$ , normalized by the applied compressive stress,  $\sigma_x^0$ , are plotted as a function of  $\Psi$  for the  $[(\pm 30)_6]_S$ ,  $[(\pm 45)_6]_S$ , and  $[(\pm 60)_6]_S$  laminates subjected to a combined shear and compression load in figures 26, 27, and 28, respectively. The addition of the shear load to the compression load causes the maximum normal stresses to increase to values approximately double the maximum normal stresses due to the compression load alone. In addition, the highest maximum normal stresses no longer occur near  $\Psi$  equal to  $0^\circ$ . Instead, they occur for values of  $\Psi$  ranging from  $20^\circ$  to  $40^\circ$ . Similar to the compression-loaded laminates, the  $[(\pm 30)_6]_S$  laminate has the highest maximum normal stresses; and, in most cases, the maximum normal stresses in the laminates under combined loading increase as  $A/D$  decreases. The maximum shear stresses,  $(\sigma_{xy})_{\text{maximum}}$ , normalized by the applied shear stress,  $\sigma_{xy}^0$ , for the laminates subjected to combined loads are shown as a function of  $\Psi$  in figures 29, 30, and 31. The addition of the compression load to the shear load increases the maximum shear stress values and changes the elliptical cutout rotation angle at which the highest maximum shear stresses occur. Similar to the shear-load-only

case, the  $[(\pm 45)_6]_S$  laminate has the highest stresses, and a decrease in A/D results in an increase in the maximum shear stress.

## CONCLUDING REMARKS

A method which combines Lekhnitskii's complex variable equations with boundary collocation and a Laurent series approximation is used to analyze the stress distributions in some finite anisotropic panels with centrally located cutouts. The maximum normal and shear stresses are found for panels with circular and elliptical cutouts and subjected to compression, shear, and combined loads. The effects of panel aspect ratio, anisotropy, cutout size, and cutout orientation on the magnitude and distribution of the stresses are examined. Angle-ply laminates, unidirectional off-axis laminates, as well as  $[(\pm 45/0/90)_3]_S$ ,  $[(\pm 45/0_2)_3]_S$ , and  $[(\pm 45/90_2)_3]_S$  laminates are studied in order to demonstrate the capabilities of the analytical method. To assess the accuracy of the method, analytical results are compared with experimental data for orthotropic panels with circular cutouts. The panels are subject to either compression or shear loading. In both cases, the analytical results show good agreement with the experimental data.

To demonstrate the flexibility of the analytical method with respect to material properties and panel geometry, several laminates subjected to compression, shear, and combined loads are analyzed. To study the effects of panel aspect ratio on laminate stresses, the limiting case of a  $(45)_{24}$  laminate with a circular cutout is analyzed and compared to more common laminates constructed of  $0^\circ$ ,  $90^\circ$ , and  $45^\circ$  plies. It is shown that the effect of panel aspect ratio on the magnitude of the stresses in panels with small cutouts is minimal, but as the cutout size increases, the effects become more significant. A study of square  $[(\pm \theta)_6]_S$  angle-ply laminates with circular cutouts identifies relationships between shear and extensional moduli and the maximum normal and shear stresses. Angle-ply laminates with high extensional moduli, such as the  $(0)_{24}$  laminate, have high maximum normal stresses when subjected to compression loading. As the fiber angle deviates from  $0^\circ$ , the maximum normal stresses decrease. Angle-ply laminates with high shear moduli, such as the  $[(\pm 45)_6]_S$  laminate, have the highest maximum shear stresses when subjected to shear loading, and as the fiber angle deviates from  $45^\circ$ , the maximum shear stresses decrease.

A study of square angle-ply laminates with elliptical cutouts whose major axes are inclined to the panel coordinate system is conducted to demonstrate the ability of the current analytical method to accommodate different cutout geometries. The results indicate that the values of the maximum stresses depend on the relationships between several geometric factors and material properties. For example, the ellipse minor-axis-to-major-axis ratio has an effect on the maximum stresses that depends on the cutout size, the fiber angle of the angle-ply laminate, and the angle of inclination of the elliptical cutout. As the inclination of the elliptical cutout is varied, the stress distribution changes dramatically, especially for shear loading where the location of the maximum shear stress also changes.

The wide range of results obtained in the present study exemplifies the ability of the present method to adapt to many different panel sizes and cutout geometries. The method allows for biaxial, shear, and combined loads. While only traction loading conditions are considered in the present paper, the method is also capable of analyzing displacement

loading conditions. The flexibility of this method characterizes its main advantage over finite elements in the areas of structural design and optimization. In addition, the FORTRAN program that implements the present method requires minimal computer time and can run interactively. Because the present method requires only two lines of input to describe the model, user modeling time is significantly less than with finite element analyses.

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Table 1. Lamina properties of Hercules, Inc. AS4/3502 graphite-epoxy material

|                                   |                     |
|-----------------------------------|---------------------|
| Longitudinal Young's Modulus, psi | $18.5 \times 10^6$  |
| Transverse Young's Modulus, psi   | $1.6 \times 10^6$   |
| Shear Modulus, psi                | $0.832 \times 10^6$ |
| Major Poisson's ratio             | 0.35                |

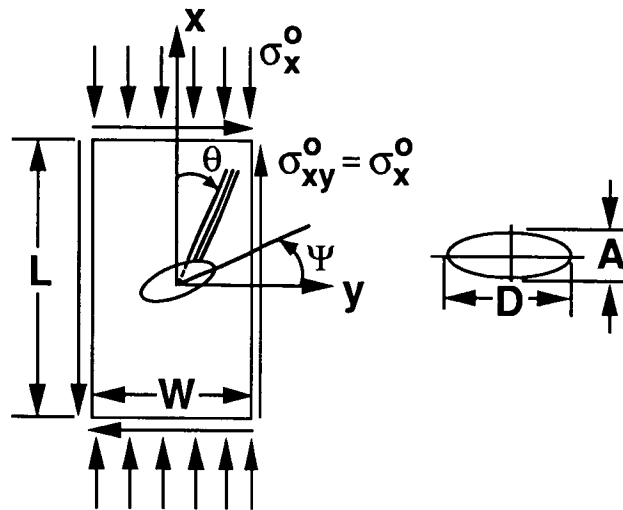


Fig. 1. Analytical model of a shear- and compression-loaded rectangular plate with a centrally located elliptical cutout inclined to the y-axis.

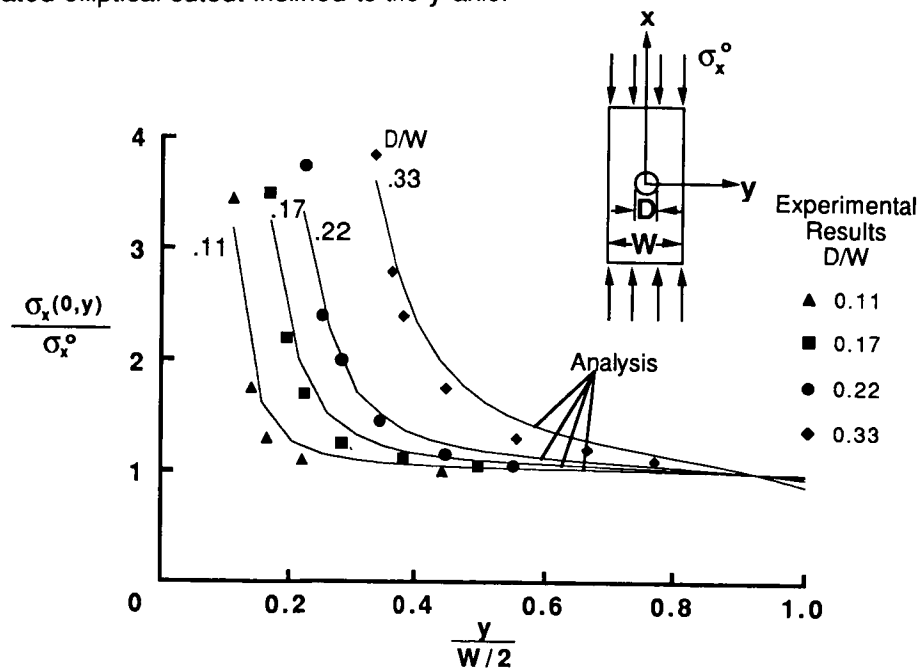


Fig. 2. Stress distribution near a circular cutout in a rectangular  $(\pm 45/0_2/\pm 45/0_2/\pm 45/0/90)_2s$  panel under applied uniaxial compression (experimental data from reference 6).



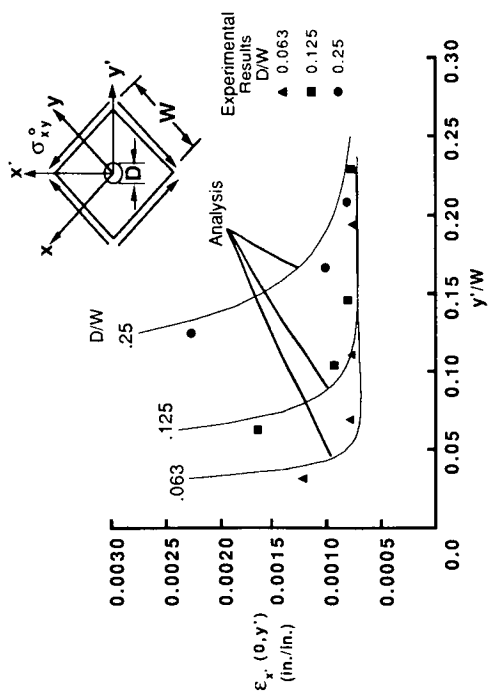


Fig. 3. Stress distribution near a circular cutout in a square  $[(\pm 45/90)_3]_s$  panel under uniform shear loading (experimental data from reference 7).

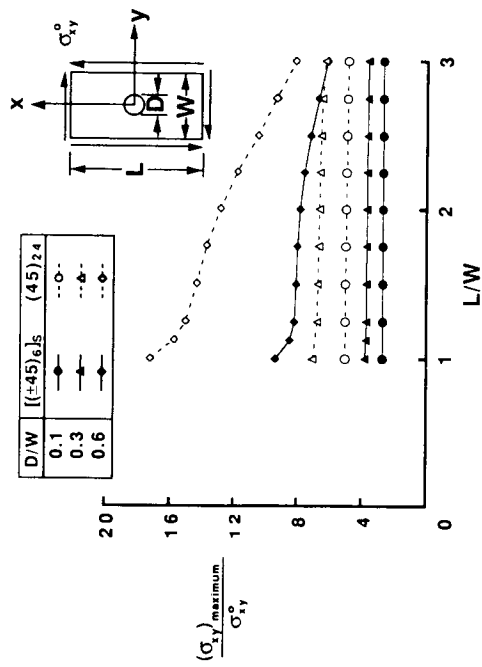


Fig. 5. Maximum shear stresses in rectangular shear-loaded  $[(\pm 45)_6]_s$  and  $(45)_{24}$  laminates with a circular cutout.

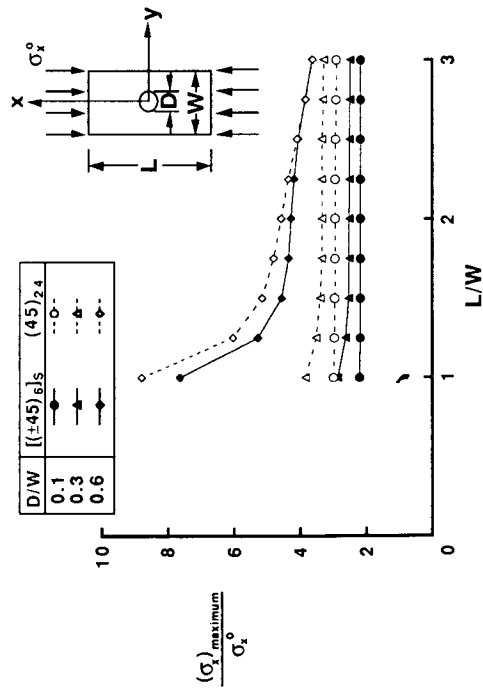


Fig. 4. Maximum normal stresses in rectangular uniaxial-compression-loaded  $[(\pm 45)_6]_s$  and  $(45)_{24}$  laminates with a circular cutout.

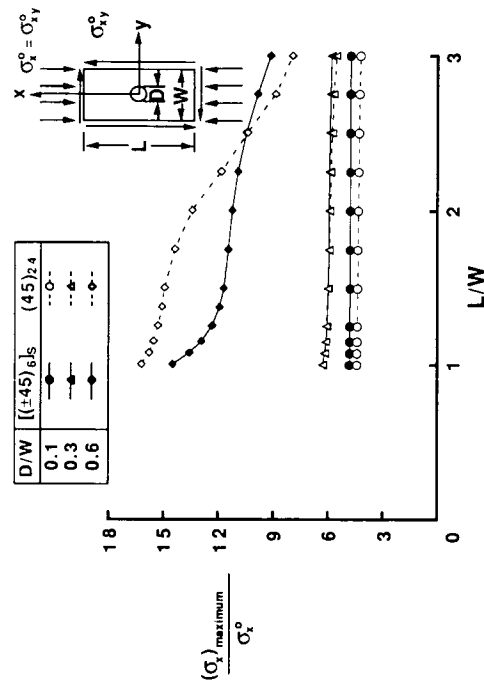


Fig. 6. Maximum normal stresses in rectangular compression- and shear-loaded  $[(\pm 45)_6]_s$  and  $(45)_{24}$  laminates with a circular cutout.

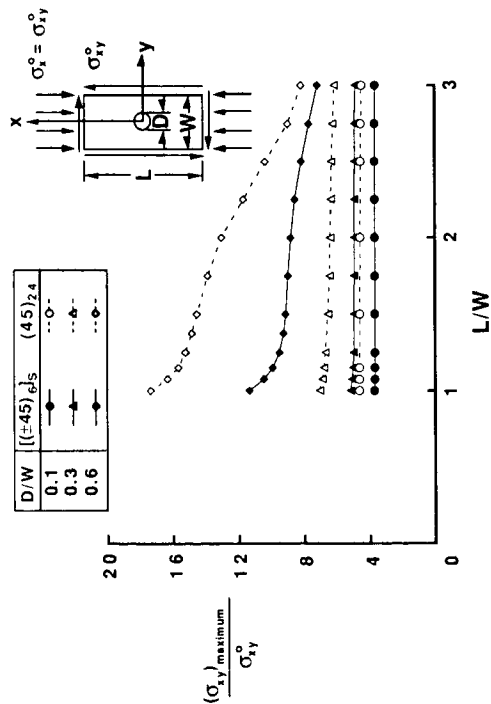


Fig. 7. Maximum shear stresses in rectangular compression- and shear-loaded [(±45)3]s and (45)24 laminates with a circular cutout.

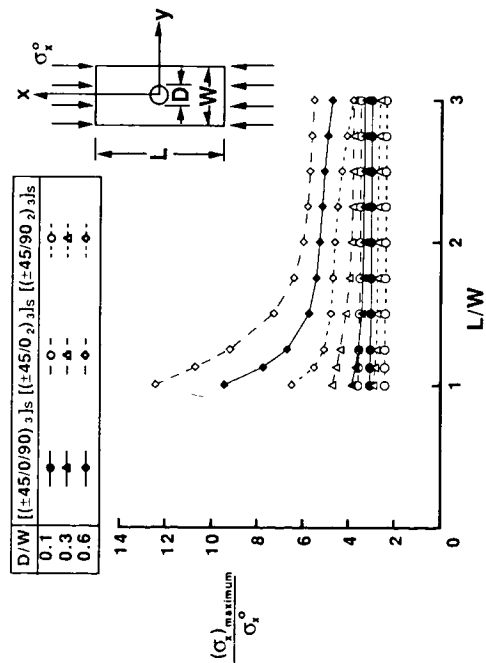


Fig. 8. Maximum normal stresses in rectangular compression-loaded [(±45/0/90)3]s, [(±45/90/2)3]s, and [(±45/90/2)3]s laminates with a circular cutout.

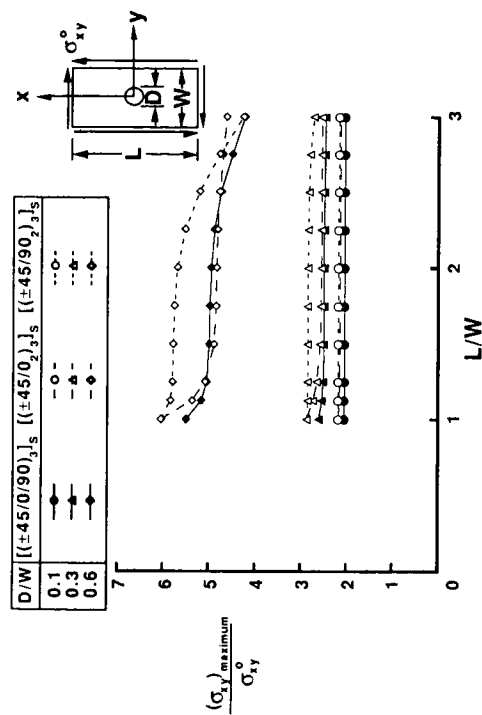


Fig. 9. Maximum shear stresses in rectangular shear-loaded [(±45/0/90)3]s, [(±45/0/2)3]s, and [(±45/90/2)3]s laminates with a circular cutout.

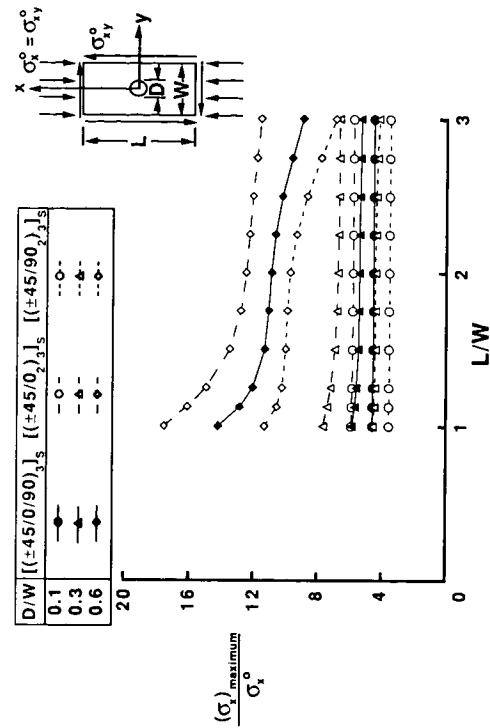


Fig. 10. Maximum normal stresses in rectangular compression- and shear-loaded [(±45/0/90)3]s, [(±45/0/2)3]s, and [(±45/90/2)3]s laminates with a circular cutout.

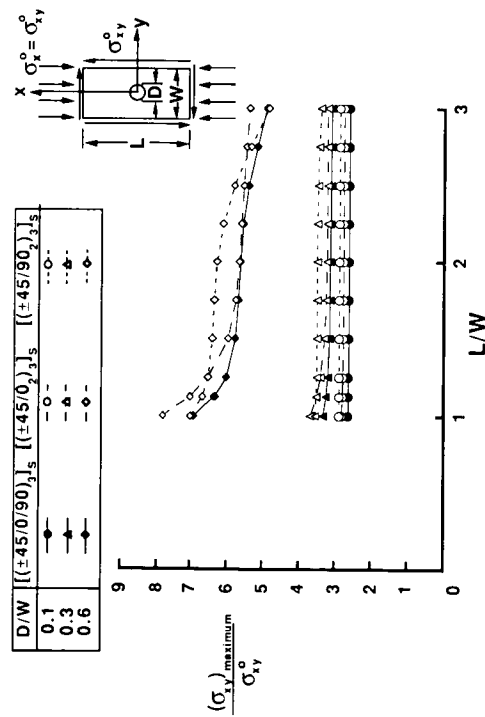


Fig. 11. Maximum shear stresses in rectangular compression- and shear-loaded  $[(\pm 45/0/90)]_s$ ,  $[(\pm 45/0)_2]_s$ , and  $[(\pm 45/90)_2]_s$  laminates with a circular cutout.

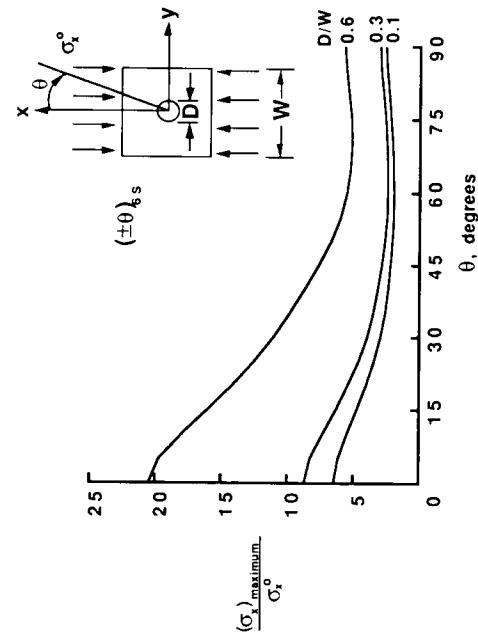


Fig. 12. Maximum normal stresses in square compression-loaded angle-ply laminates with a circular cutout.

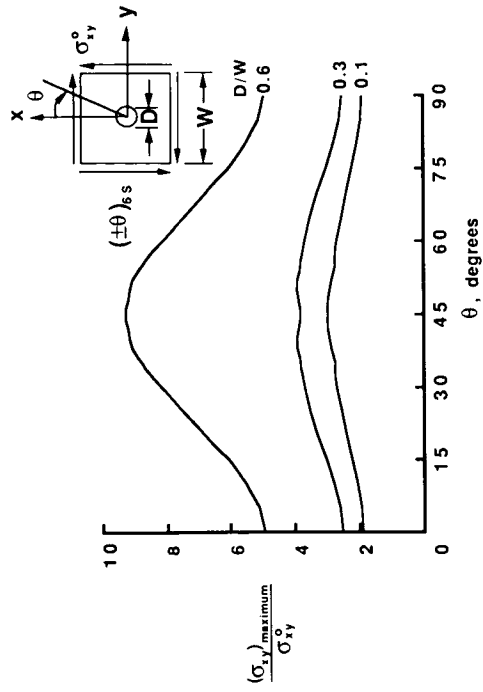


Fig. 13. Maximum shear stresses in square shear-loaded angle-ply laminates with a circular cutout.

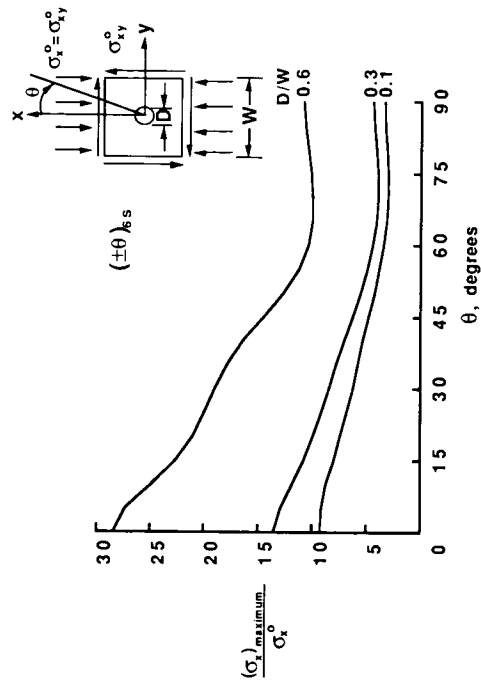


Fig. 14. Maximum normal stresses in square compression- and shear-loaded angle-ply laminates with a circular cutout.

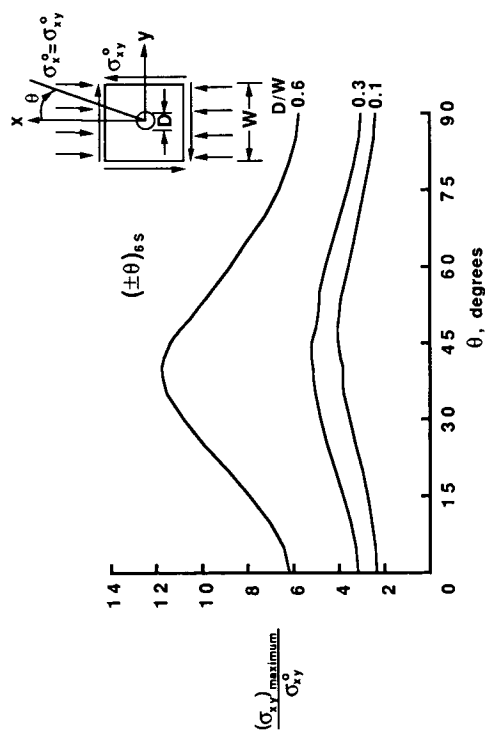


Fig. 15. Maximum shear stresses in square compression- and shear-loaded angle-ply laminates with a circular cutout.

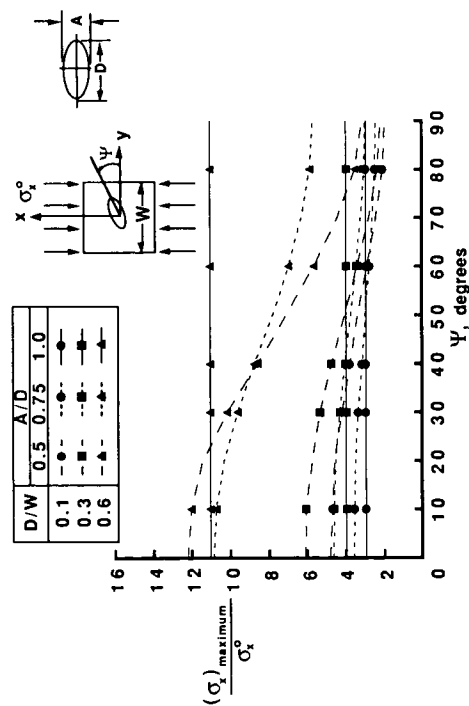


Fig. 16. Maximum normal stresses in a square compression-loaded  $[(\pm 30)_6]_s$  laminate with an elliptical cutout inclined to the  $y$ -axis.

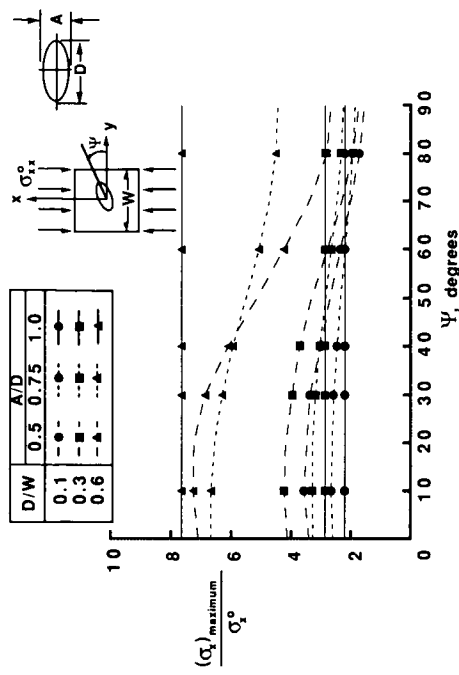


Fig. 17. Maximum normal stresses in a square compression-loaded  $[(\pm 45)_6]_s$  laminate with an elliptical cutout inclined to the  $y$ -axis.

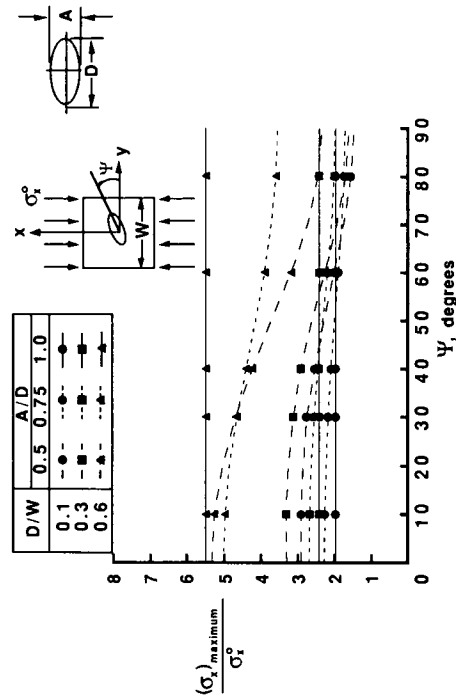


Fig. 18. Maximum normal stresses in a square compression-loaded  $[(\pm 60)_6]_s$  laminate with an elliptical cutout inclined to the  $y$ -axis.

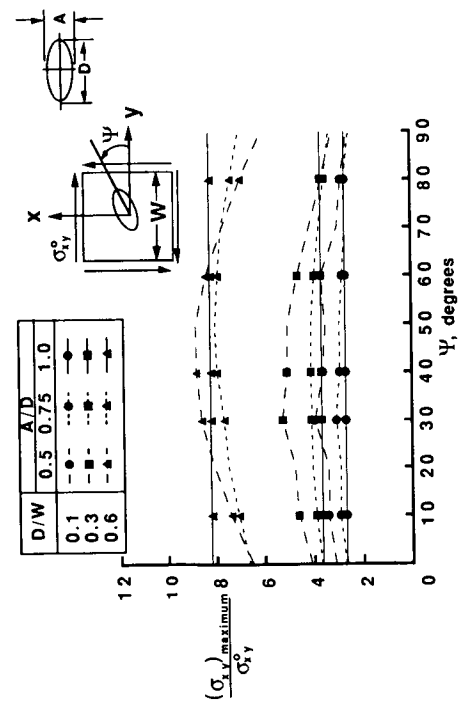


Fig. 19. Maximum shear stresses in a square shear-loaded  $[(\pm 30)_6]_s$  laminate with an elliptical cutout inclined to the y-axis.

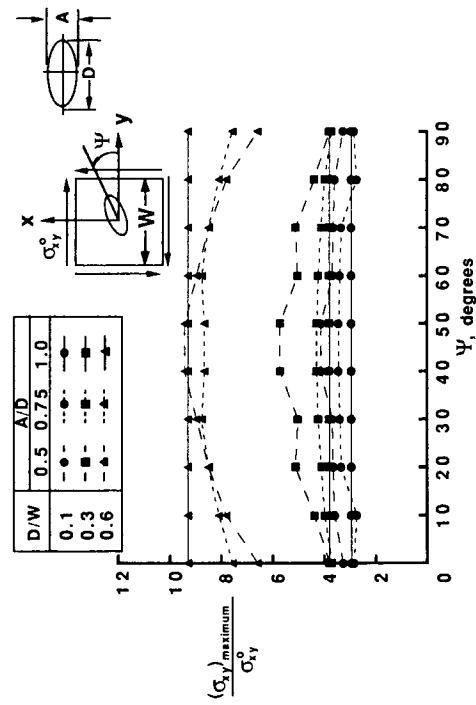


Fig. 20. Maximum shear stresses in a square shear-loaded  $[(\pm 45)_6]_s$  laminate with an elliptical cutout inclined to the y-axis.

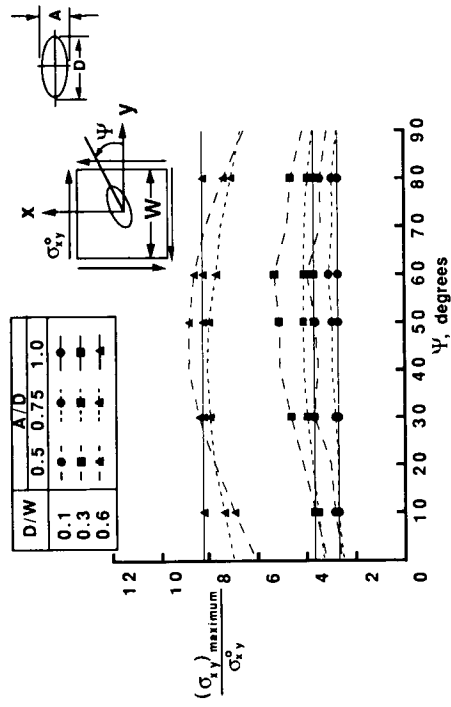


Fig. 21. Maximum shear stresses in a square shear-loaded  $[(\pm 60)_6]_s$  laminate with an elliptical cutout inclined to the y-axis.

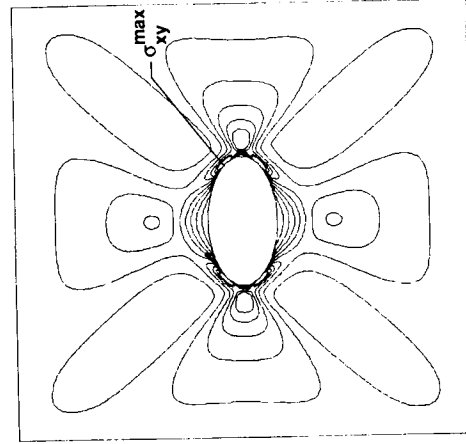


Fig. 22. Shear stress distribution for a square shear-loaded  $[(\pm 45)_6]_s$  laminate with an elliptical cutout aligned with the y-axis.

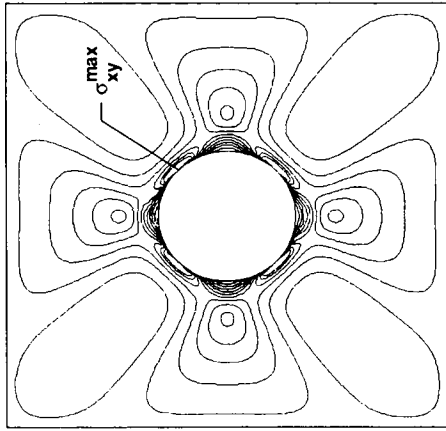


Fig. 25. Shear stress distribution for a square shear-loaded  $[(\pm 45)_6]_s$  laminate with a circular cutout.

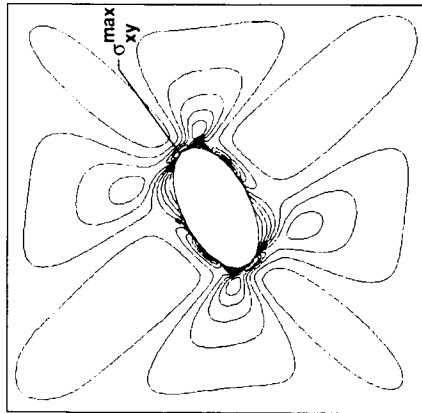


Fig. 23. Shear stress distribution for a square shear-loaded  $[(\pm 45)_6]_s$  laminate with an elliptical cutout inclined at  $25^\circ$  to the y-axis.

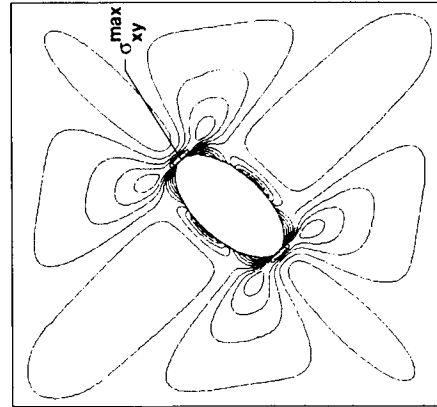


Fig. 24. Shear stress distribution for a square shear-loaded  $[(\pm 45)_6]_s$  laminate with an elliptical cutout inclined at  $45^\circ$  to the y-axis.

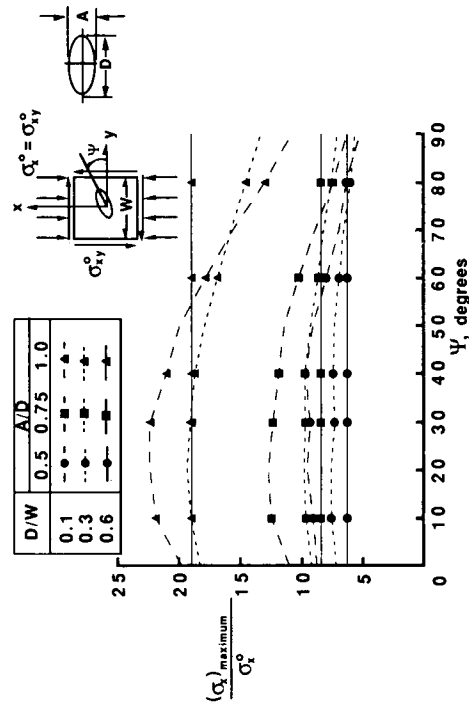


Fig. 26. Maximum normal stresses in a square compression- and shear-loaded  $[(\pm 30)_6]_s$  laminate with an elliptical cutout inclined to the y-axis.

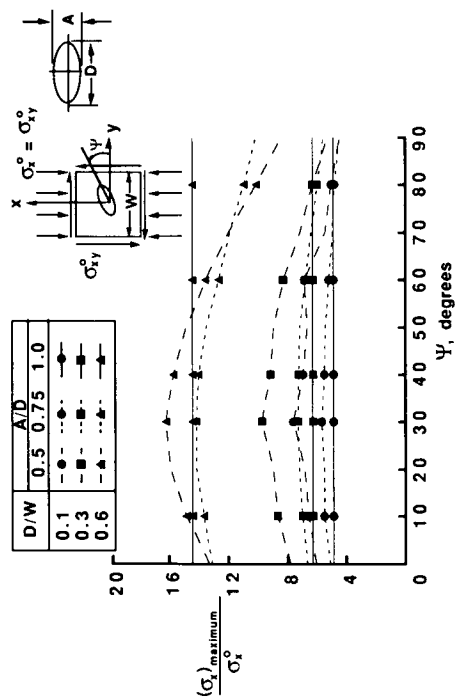


Fig. 27. Maximum normal stresses in a square compression- and shear-loaded  $[(\pm 45)_2]_s$  laminate with an elliptical cutout inclined to the y-axis.

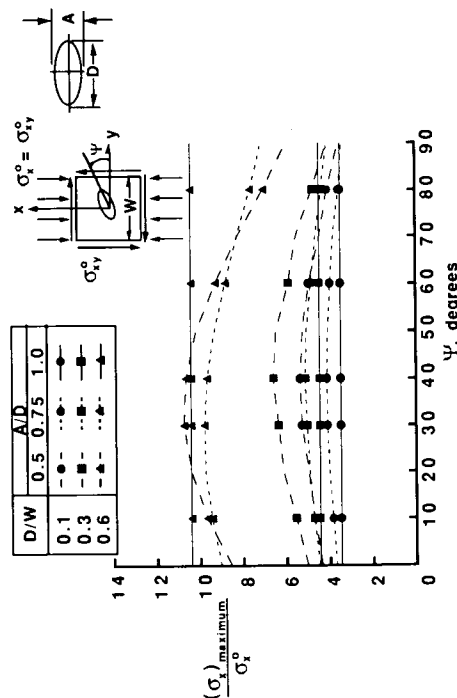


Fig. 28. Maximum normal stresses in a square compression- and shear-loaded  $[(\pm 60)_2]_s$  laminate with an elliptical cutout inclined to the y-axis.

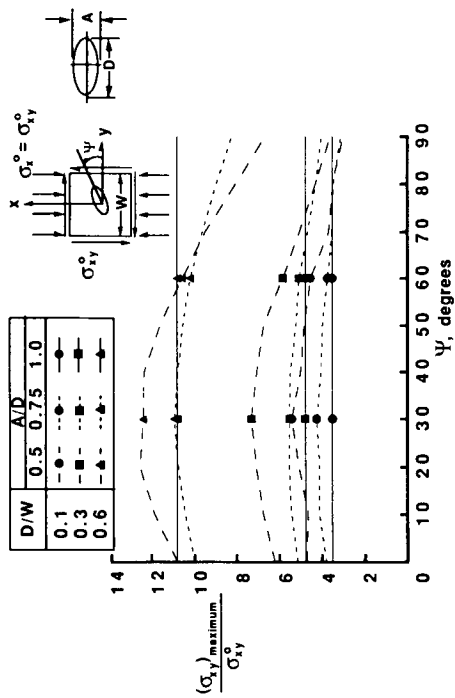


Fig. 29. Maximum shear stresses in a square compression- and shear-loaded  $[(\pm 30)_2]_s$  laminate with an elliptical cutout inclined to the y-axis.

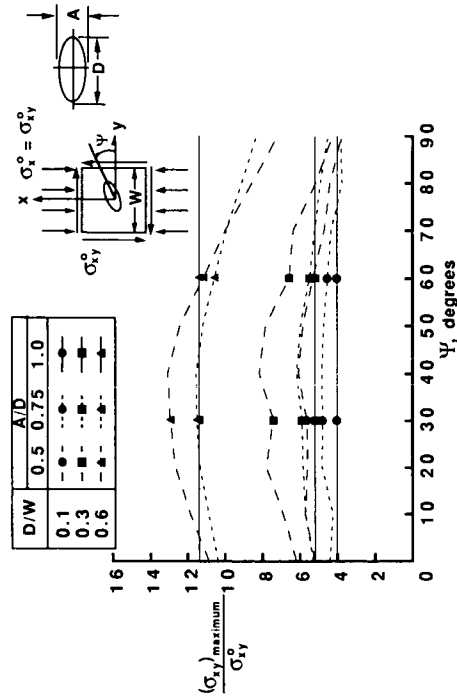


Fig. 30. Maximum shear stresses in a square compression- and shear-loaded  $[(\pm 45)_2]_s$  laminate with an elliptical cutout inclined to the y-axis.

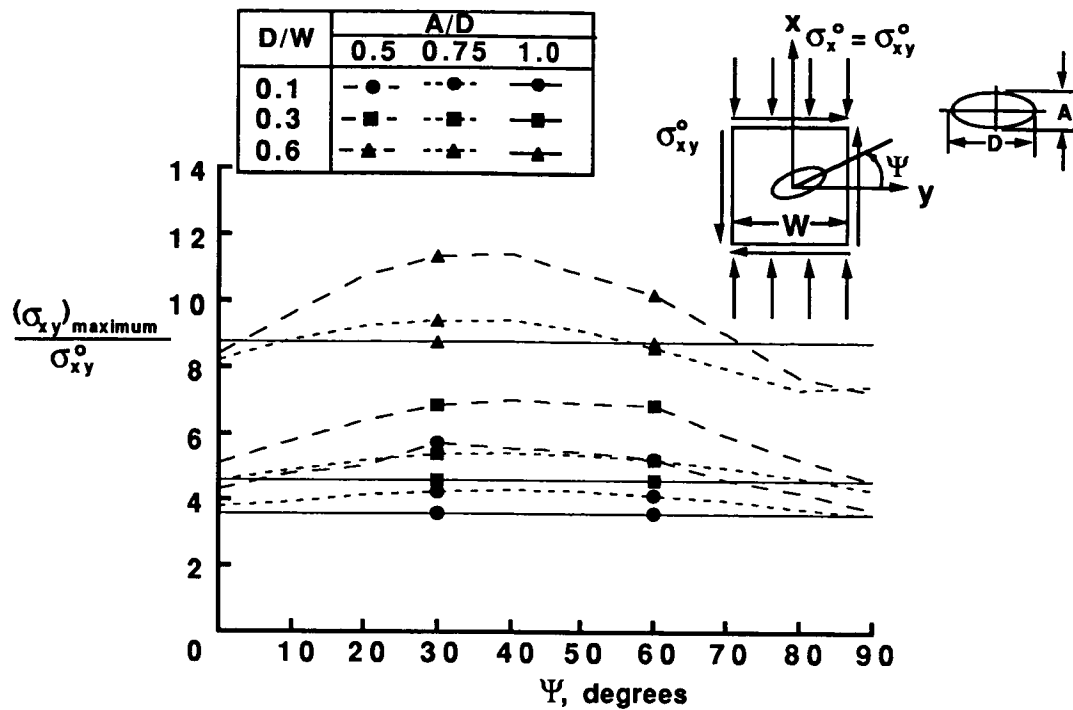


Fig. 31. Maximum shear stresses in a square compression- and shear-loaded  $[(\pm 60)_6]_S$  laminate with an elliptical cutout inclined to the  $y$ -axis.



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# THE ROLE OF BIAXIAL STRESSES IN DISCRIMINATING BETWEEN MEANINGFUL AND ILLUSORY COMPOSITE FAILURE THEORIES

by

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## ABSTRACT

The irrelevance of most composite failure criteria to conventional fiber-polymer composites is claimed to have remained undetected primarily because the experiments that can either validate or disprove them are difficult to perform. Uniaxial tests are considered inherently incapable of validating or refuting *any* composite failure theory because so much of the total load is carried by the fibers aligned in the direction of the load. The Ten-Percent Rule, a simple rule-of-mixtures analysis method, is said to work well *only* because of this phenomenon. It is stated that failure criteria can be verified for fibrous composites only by biaxial tests, with orthogonal in-plane stresses of the same as well as different signs, because these particular states of combined stress reveal substantial differences between the predictions of laminate strength made by various theories. Three scientifically plausible failure models for fibrous composites are compared, and it is shown that only the in-plane shear test (orthogonal tension and compression) is capable of distinguishing between them. This is because most theories are "calibrated" against the measured uniaxial tension and compression tests and any cross-ply laminate tests dominated by those same states of stress must inevitably "confirm" the theory.

## BACKGROUND

For several years, the author has tried to expose and rectify serious fundamental deficiencies in the most widely taught "failure theories" for composite laminates. This has proved to be a most difficult task, mainly because of a widespread reluctance to use any method not already coded on a computer or to challenge any output from the computer. The difficulty of generating valid test data with which to accept or reject any theory is also a factor.

The issue of computer codes for the new theory is being addressed in another paper. The emphasis here is on the need to validate theories by tests on structural laminates, particularly under biaxial in-plane loads. Unfortunately, there is only a limited appreciation of difficulties with the design and execution of even the standard uniaxial tests on cross-ply laminates. These problems are exacerbated by a failure to recognize that gross oversimplifications have been made in the model used to formulate most composite failure criteria. Consequently, most biaxial test coupons fail prematurely outside the test section in areas of uniaxial stress because the target biaxial strengths have been badly underestimated.

Some promoters of abstract mathematical failure models for composite materials have taken advantage of these experimental difficulties. They do not find it necessary to conduct experiments at the structural *laminate* level to validate predictions made by theory. Instead, they characterize the material by a series of tests at the *lamina* coupon level, relying on a theory with sufficient adjustable and sometimes unmeasurable parameters to match the lamina test results. Since no *mathematical* approximations are made in the derivations, it is implied that there is no need for further tests which, if conducted for a sufficiently wide range of stress states, might expose inconsistencies in the predictions when the model used for the theory does not represent physical reality. Conversely, redundant testing would inevitably validate a scientifically sound theory unless the experiments

were faulty. If 10 successful tests were run to deduce eight unknown properties, the same properties should be predicted no matter which eight tests were selected for the analysis. The scheme of ensuring that redundant tests will not be conducted so that one's theory can *never* be challenged has also been used in bolted composite joint studies. The technique has been applied so artfully that few would question it.

Such thinking has not only led most composite structural analysts to refrain from questioning the foundations of their computer codes for strength prediction, but has also deluded some who would apply theories of anisotropic elasticity for *homogeneous* materials to distinctly *heterogeneous* fiber-polymer composites into believing that there is nothing wrong with such a "simplifying" assumption, merely because the individual fibers in the composite are microscopic.

Others have justified the need for such simplifications by the complexity of micromechanics, suggesting that *some* simpler theory had to be developed for all those who would not use *any* theory far more complicated than what they used for metallic structures. This problem is not helped by the widespread use of finite-element procedures without ensuring that both the model and the boundary conditions simulate reality. But, to be fair, the most potent argument against micromechanical analyses is the large number of material properties that cannot be easily determined experimentally but are needed to implement the more realistic failure models.

The end result of all this is that few if any composite failure theories have *ever* been properly verified by experiment.

This by no means implies that all composite structures designed and built so far are unsafe; typically, less than 1 percent of composite structures on large aircraft is actually governed by *unnotched* laminate strengths. The rest is dominated by joints, damage tolerance, and stiffness requirements. Empirical interpretations of data are needed for joint strength and damage tolerance, while the laminate stiffnesses are not in doubt because lamination theory works for even heterogeneous materials. At least the predicted elastic constants are right, even if the strengths are wrong.

Further, nearly all composite structures built so far have been certified by test rather than by analysis. And things are likely to stay this way unless better, more realistic theories are developed.

### HOW MANY MEASUREMENTS ARE NEEDED TO CHARACTERIZE THE STRENGTH OF A FIBROUS COMPOSITE LAMINATE?

Considerable confusion exists as to the number of measurements needed to characterize the strength of composite materials. The seemingly reasonable position that it is necessary to characterize longitudinal and transverse strengths, in both the tensile and compressive directions, and to have some measure of the shear strength would suggest that five measurements are needed. Many composite failure models have been based on such an assumption, adopted because of the apparent ease with which those particular measurements could be made. But if one were to consider the real physics of the situation instead, one would conclude that only *one* measurement is needed to characterize *each mode* of failure. If the same mode of failure, such as yielding in ductile metal alloys, occurred under different states of combined stress, measurements made under different states of stress would be *equivalent* and not independent. The value of redundant tests would be to demonstrate a consistency confirming that the theory was sound.

Thus, the five strength measurements would be appropriate *only* if there were precisely five modes of failure to be characterized. And, *under no circumstances* could these five measurements be integrated into a *single* smooth, continuous failure envelope. They should represent five *superimposed* envelopes, truncating each other locally so that one or another would govern as the state of stress varied.

It is clear, then, that the unstated simplifying assumptions of traditional composite failure theories are so contradictory to basic laws of physics that the theories should be discarded. However, it is commonly held that any new and better theory must inevitably be *more complicated* than older theories and need additional data for its implementation. A claim that an entire failure envelope can be constructed from a *single* test result and simultaneously be more accurate than older theories based on measurements of *four* distinct measurements of strength seems difficult to accept, even when it is explained that additional failure modes can be covered, if needed, at the rate of one test per failure mechanism. The conclusion seems to run contrary to tradition: the new theory needs, at most, measurements of the longitudinal strength of the lamina in tension and compression, whereas older theories needed transverse strengths as well, and the *omission* of the transverse measurements *improves* the accuracy of the theory.\*

Separate failure envelopes for shear failures of the fibers and in-plane shear failures for the matrix can be superimposed at the lamina level, as shown in Figure 1. Their origins are offset to account for residual thermal stresses induced by curing at elevated temperatures. Apart from traditional elastic constants such as Young's moduli and the various Poisson's ratios, Figure 1 would be based on *only two* measurements of strength, one for the fibers and one for the matrix.

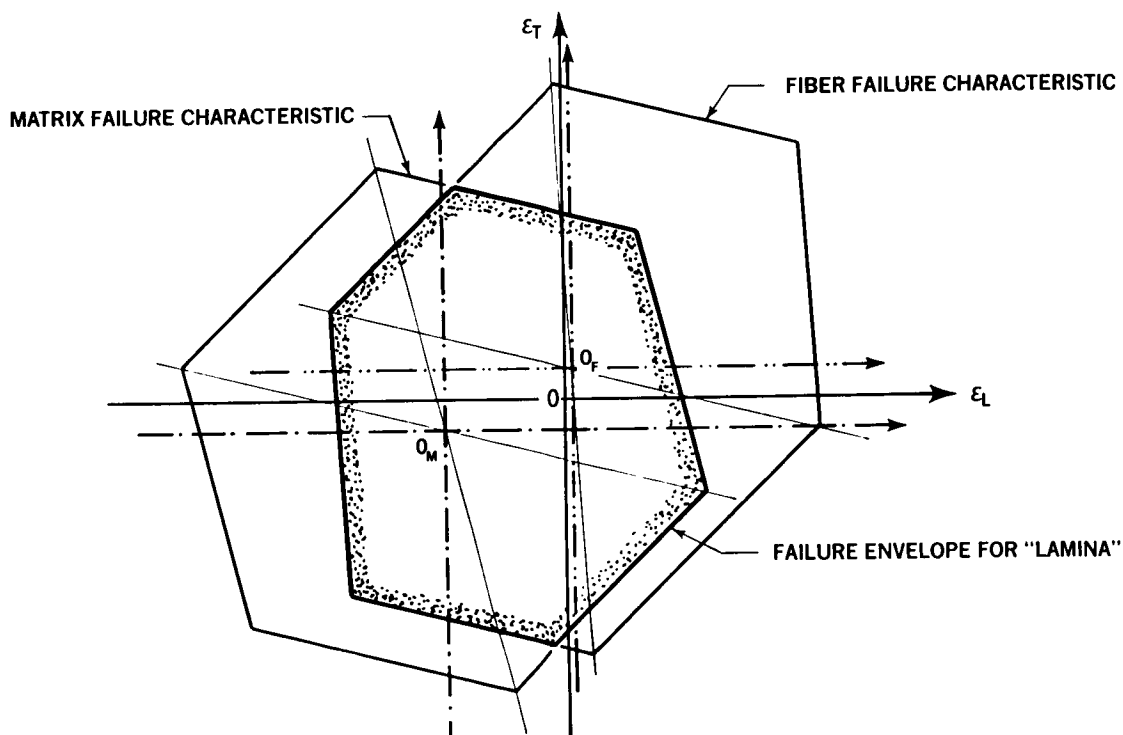


FIGURE 1. SEPARATE FAILURE CHARACTERISTICS FOR FIBERS AND MATRIX

### THE INVALIDITY OF ASSOCIATING COMPOSITE FAILURE CRITERIA WITH HILL'S WORK ON PLASTICITY IN METALS

Many composite failure theories have been falsely likened to the *unrelated* work of Hill (Reference 2) on the plastic yielding of slightly anisotropic ductile metals. The very name Tsai-Hill for one of the early composite

\* The transverse-tension strength measured on a unidirectional lamina has no relevance to any in-plane strength of a multidirectional structural laminate, whether the failure be matrix-dominated or fiber-dominated. However, an empirically deduced "effective" transverse-tension strength can be used to provide a *separate* failure characteristic truncating the fiber-failure envelope locally throughout part of the tension-tension stress quadrant (see Reference 1.)

failure theories (Reference 3) implies such an association. In fact, there is absolutely *no similarity* between the two situations. Hill's theory of plasticity characterizes the yielding of a *homogeneous* material under various states of combined stress by a *single* mechanism, while the other theory and its innumerable clones refer to failures by at least four and sometimes five different mechanisms of a distinctly *heterogeneous* composite.

If Hill had tried to adapt his own methods to predicting the strength of fibrous composite laminates, he would most likely have developed a *separate* curve for each possible failure mechanism. One or another mechanism would prevail, depending on the state of combined stresses, and the failure "envelope" would have been kinked wherever the failure mechanism changed. Instead, today, industry and academia alike have what can only be described as a plethora of *meaningless smooth curves passed through unrelated data points* as the "characterization" of unidirectional composites. These curves then serve as the basis for predicting the strength of cross-ply composite laminates.

This misunderstanding is highlighted in Figure 2, in a different context, to illustrate the absurdity of the conventional composite failure model shown in Figure 3. Figure 4 shows a further variation of this theme. In every case, some meaning can be ascribed to the axes themselves, no matter how difficult they may be to measure: the problem is in interpreting intermediate points, as indicated by the question mark in each of the figures.

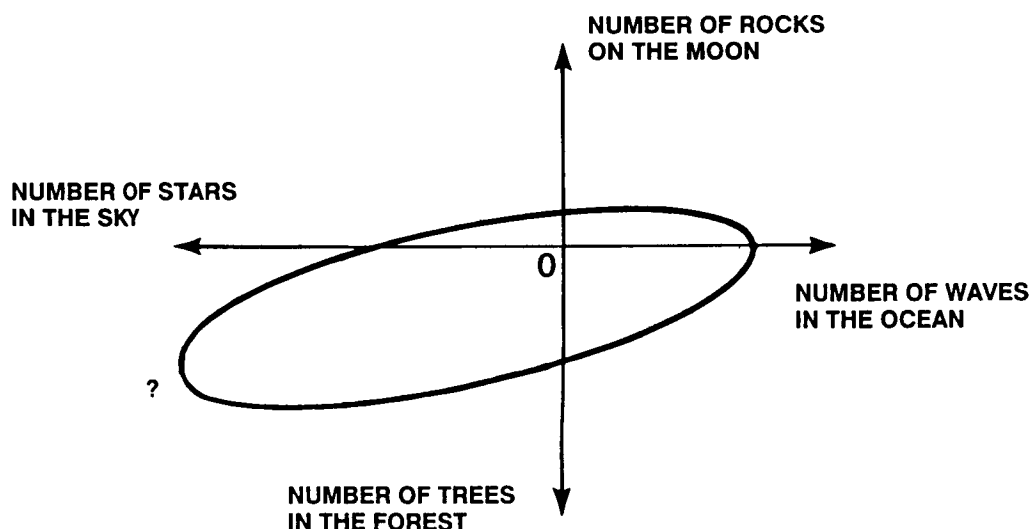


FIGURE 2. ONE EXAMPLE OF A MEANINGLESS CURVE DRAWN THROUGH UNRELATED DATA POINTS

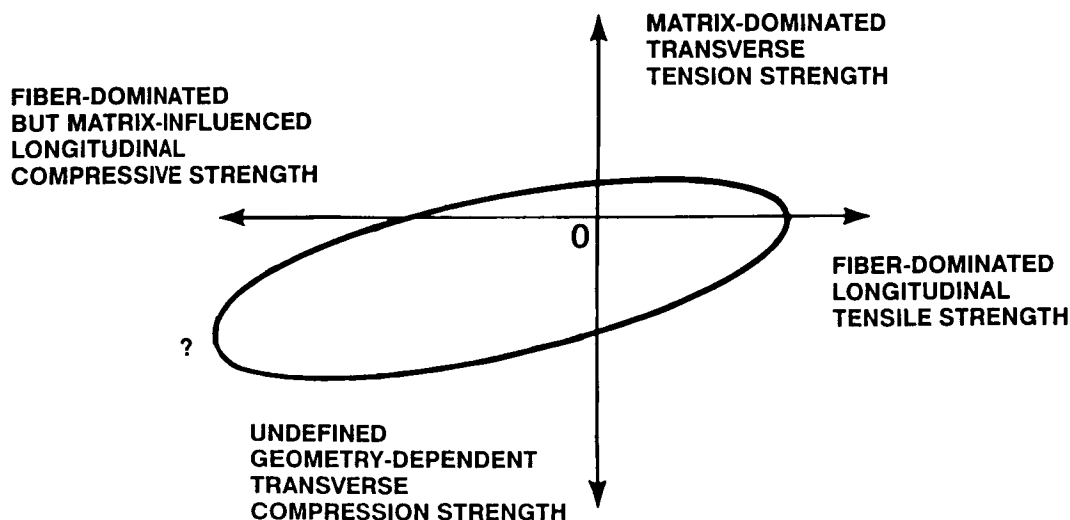
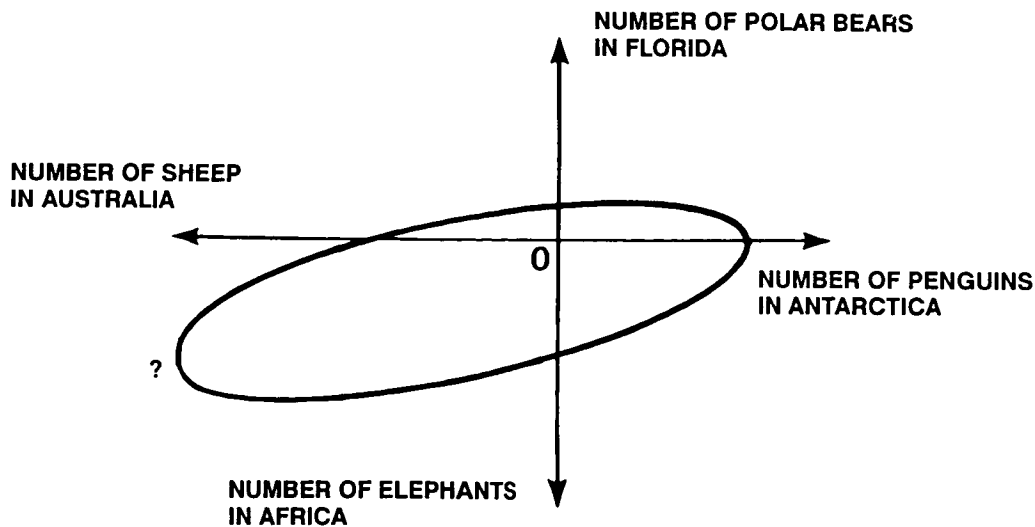


FIGURE 3. AN EQUALLY MEANINGLESS CURVE DRAWN THROUGH UNRELATED DATA POINTS



**FIGURE 4. ANOTHER EXAMPLE OF A MEANINGLESS CURVE DRAWN THROUGH UNRELATED DATA POINTS**

Hill's use of *three* parameters to characterize his failure surface for mildly anisotropic materials may seem to contradict the author's assertion that only one should be needed when only one failure mode is involved. However, while it takes only two points to specify a straight line, the line may be specified by three or more points, provided that they are merely redundant and not contradictory. The Tresca and von Mises' criteria for ductile materials require only *one* parameter each to characterize the yield of a ductile material under *any* set of combined stresses. Logic suggests that, if one were to perform the necessary algebra, one could also deduce an equivalent *single* parameter for orthotropic materials, provided that only one failure mechanism was involved. The other two parameters would be replaced by elastic constants characterizing the degree of anisotropy. Hill found a way of circumventing such tedious work. In all probability, he arrived at a far more elegant expression of essentially the same result, or an extremely close approximation to it.

If three distinct modes of failure had been involved for different states of stress — yielding under shear, short-transverse-grain delaminations, and brittle fracture, for example — no *single* characterization of the strength would be possible. But neither would Hill's three parameters suffice since they would merely represent the redundant specification of *one* mode of failure. These three different modes of failure would certainly be associated with three different strengths along the principal material axes, but that is the end of the similarity. Their characterization would require *three* physically independent parameters and the failure surface would certainly *not* be smooth.

Rowlands (Reference 4), in an excellent summary of the history of composite strength failure theories, reveals the extent of misinterpretation of Hill's work. He states that "Hill's theory was adapted by Azzi and Tsai as a strength criterion for composites" (p. 76), and later states "While it is not common to use Equation (43) [one particular formulation of Hill's theory] with composites, this concept does form the basis of several composite strength criteria" (p. 90). But, later on the same page he states that "Unlike the maximum stress or strain criteria, Equation (43) contains interaction among the stresses and *therefore involves combined modes of failure*," [emphasis added]. Rowlands also states (pp. 90 and 91) that "This led Hill (1950) to propose *an* orthotropic yield condition" [emphasis added]. How does this imply "combined modes of failure?"

*Nothing* in Hill's work addresses more than one mode of failure and he should therefore be spared the ignominy of association with the many abstract mathematical failure theories for composite materials. Yet, in the United Kingdom and Europe, Tsai's misinterpretation of Hill's theory of anisotropic plasticity is referred to as the "modified Hill theory."

Earlier, Rowlands states (p. 86) that "Yielding normally does not occur in fiber-reinforced plastics in the same sense as in metals. Nevertheless, many of the orthotropic strength theories are anisotropic extensions of iso-

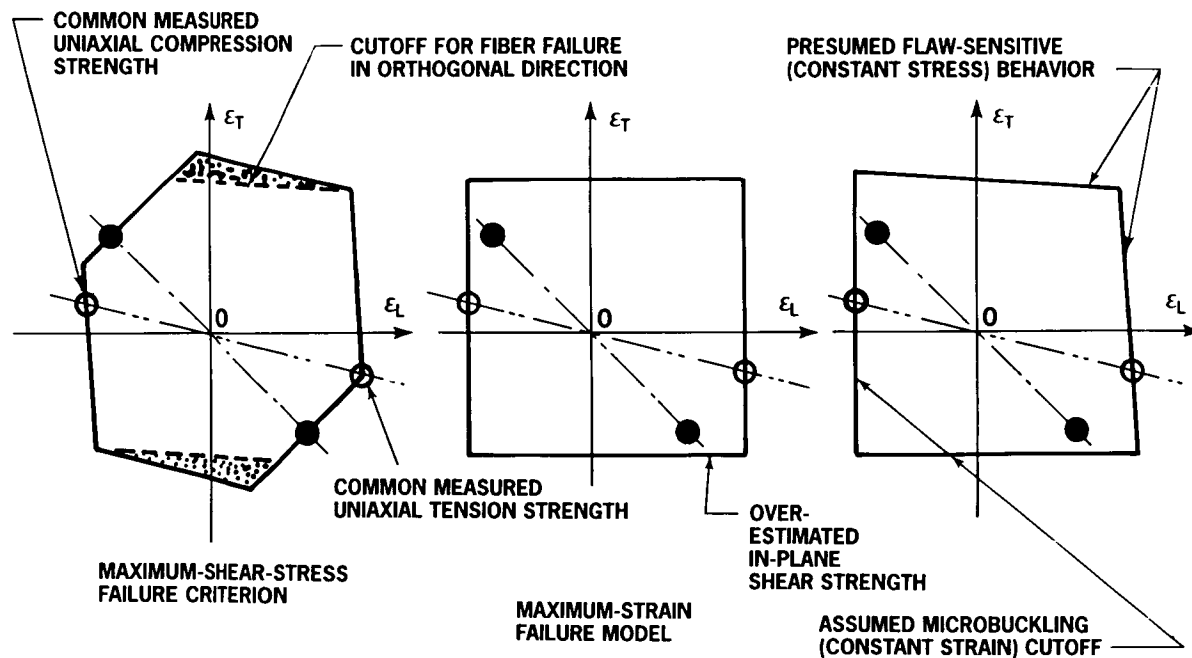
tropic yield criteria." He also states (p. 96), "In an effort to more accurately predict experimental results Tsai and Wu (1971) proposed a lamina failure criterion having additional stress terms not appearing in theories such as the Hill analysis."

What is involved is a simple curve-fit, which has no association with the physics of the situation. Additional tests are needed to provide data for each additional term included in the theoretical failure model. Ironically, as is well known, the Tsai-Wu failure model (Reference 5) contains one interaction term for which no reliable measurement has been found. Instead, it is customarily assigned the value 0.5 or zero.

## THE NEED FOR BIAXIAL RATHER THAN UNIAXIAL LAMINATE TESTS TO VALIDATE FAILURE THEORIES

Unfortunately, the characteristics of fiber-polymer composites in which strong, stiff fibers are embedded in relatively soft matrices are such that "failure theories" can *never* be validated, or even repudiated, by uniaxial testing alone. Indeed, the author's Ten-Percent Rule (Reference 6) for preliminary design by mental arithmetic works as well as it does *only* because of the dominance of the load carried by fibers aligned with the applied load. Only the *biaxial* strengths of cross-plyed laminates provide a means of differentiating between good and bad methods. And it transpires that at least *two* theories can be validated if attention is confined to only those biaxial in-plane stresses in which the stress components have the *same* sign. Of all the possible states of stress with which to assess composite failure theories, the in-plane shear state is the most crucial. But a theory cannot be validated without also considering biaxial stresses of the same sign.

The criticality of the in-plane-shear state of stress in differentiating between plausible and implausible failure models is explained in Figure 5. Except for the tension-compression (shear) quadrants, virtually the same composite laminate strengths would be predicted by the author's generalized maximum-shear-stress failure



NOTE: ONLY THE IN-PLANE SHEAR TEST CAN DISTINGUISH BETWEEN THESE PHYSICALLY DIFFERENT FAILURE MODELS SINCE THE UNIAXIAL TENSION AND COMPRESSION STRENGTHS (OPEN SYMBOLS) ARE COMMON THROUGHOUT

FIGURE 5. IMPORTANCE OF BIAXIAL TESTING TO DISTINGUISH BETWEEN FAILURE MECHANISMS

criterion, the maximum-strain model, and a combination of flaw-sensitive fracture in tension and some form of instability in compression. The other predicted strengths are similar because all three models are empirically forced to pass through the *same* measured tensile and compressive strengths under uniaxial loads. The theories predict different strengths only under in-plane shear loads, so that is the *only* test capable of validating or repudiating any of these proposed failure mechanisms.

On the other hand, agreement between test and the predictions in all the stress quadrants using the author's failure model does not actually prove that flaw-sensitive fracture could not occur under tensile loads alone. All that can be said with certainty is that the other models cannot possibly be valid throughout *all* states of combined stresses.

## USE OF MATRIX "FAILURES" TO TRUNCATE PREDICTED FIBER-DOMINATED STRENGTHS

A number of theories postulate that the maximum-strain theory is valid for the fibers but that it sometimes needs truncating to allow for matrix-dominated failures. This concept appears to result from a perception that the fiber-based maximum-strain theory is in such close agreement with tests for some states of stress that it must be a valid basis for a composite failure criterion, even if it does need some minor adjustment for other states of stress.

Perhaps the best known of these works on failure criteria with multiple characterizations is by Puck (Reference 7), who has influenced others to follow his rationale. In one respect, he is quite correct in separating the failure criteria for the fibers and the resin, although he seems to have been unaware that he could not possibly characterize the state of stress in the resin with a simple theory that does not provide for residual curing stresses. Similarly, more complex treatments such as those of Grant and Sanders (Reference 8) have also relied on presumed matrix failures to modify the maximum-strain failure model for shear- and compression-dominated loads.

As with much of Tsai's work, use of postulated matrix failures to truncate a fiber-failure envelope seems quite plausible. And, under other circumstances, such truncations are undoubtedly true. However, in this specific case, the cutoffs are not consistent with other test data. The very highest measured in-plane shear strength of an all  $\pm 45^\circ$  laminate has the fibers failing at barely *half* the axial strain at which they fail under uniaxial loads. This implies that the matrix strains are also barely half as high, which leads to the following question: How can a matrix failure be used to explain a fiber-dominated in-plane shear strength when both the matrix and the fiber can withstand *twice* as much load under uniaxial tension or compression?

Researchers such as Puck, and Grant and Sanders must have been aware that the fiber-dominated maximum-strain failure model predicted excessive strengths for in-plane shear-dominated loads since their "matrix" failure equations truncated those regions — and sometimes the compression-dominated regions — without changing predictions for the tension-tension quadrant. It seems strange that these authors and others did not accept the same fiber-dominated maximum-strain failure model as a basis for a complete failure criterion and modify the *fiber* failure criterion for those states of stress for which the model was inadequate. This would have produced an even better theory and avoided the need to introduce many other experimentally determined reference strengths for the theory.

The truncated maximum-strain theory proposed in Reference 9 is almost indistinguishable from the author's generalized maximum-shear-stress failure criterion for orthotropic materials such as carbon fibers, and differs from the original untruncated maximum-strain model only for in-plane shear-dominated loads. This approach limits predicted in-plane shear strengths just as effectively as matrix cutoffs, but without doubling the number of input properties needed for the analysis. In any case, it is more in keeping with the physics of the problem.



Curiously, work by Grimes and others at Northrop (Reference 10) included a cutoff with similar consequences but for entirely different reasons. Grimes imposed a limit on the shear deformation a resin matrix could withstand if there were no fibers in some direction to restrict the strain. He set the limit in the form of a shear strain between the  $0^\circ$  and  $90^\circ$  directions for either a woven or unidirectional layer of composite material. He intended to confine the matrix to elastic deformations only: the shear strain limit was set slightly higher than that associated with the unidirectional tension state of stress, so as not to undercut that seemingly valid test result. But, although he was not aware of this at the time, the in-plane shear cutoff also *implied* a limit on the axial strains of any fibers in the  $\pm 45^\circ$  directions to far below the fiber strains which the maximum-strain failure model would have permitted. Indeed, Grimes's cutoff has virtually the same effect on predicted fiber-dominated strengths in the tension-compression quadrant as the author's own theory. And to think that it all originated from a desire to prevent the matrix from cracking! Nevertheless, the linear limit on design shear strains in the matrix implies that the actual ultimate failure would occur at higher loads. There is no evidence to support this.

In the late 1960s, long before Grimes's work was published, entirely empirical truncations for in-plane shear strengths were made at Grumman Aircraft on the lamina rather than on the fiber or matrix, to achieve a similar end (see Reference 11).

These works, as well as that by Black (Reference 12), shown in Figure 6, are noteworthy because they imply (1) an acknowledgment that classical composite failure theory is inferior to empirical modifications of the popular maximum-strain failure model for fiber-polymer composites, and (2) an acceptance of predicted strengths similar to those later predicted by the author using a generalization of the maximum-shear-stress failure criterion as his physical model.

Despite a paucity of publications on this topic and a lack of agreement on a single failure model, the aerospace industry has found reliable empirical techniques to predict the strength of composite laminates quite independently of the neoclassical mathematical theories of anisotropic elasticity for truly homogeneous materials.

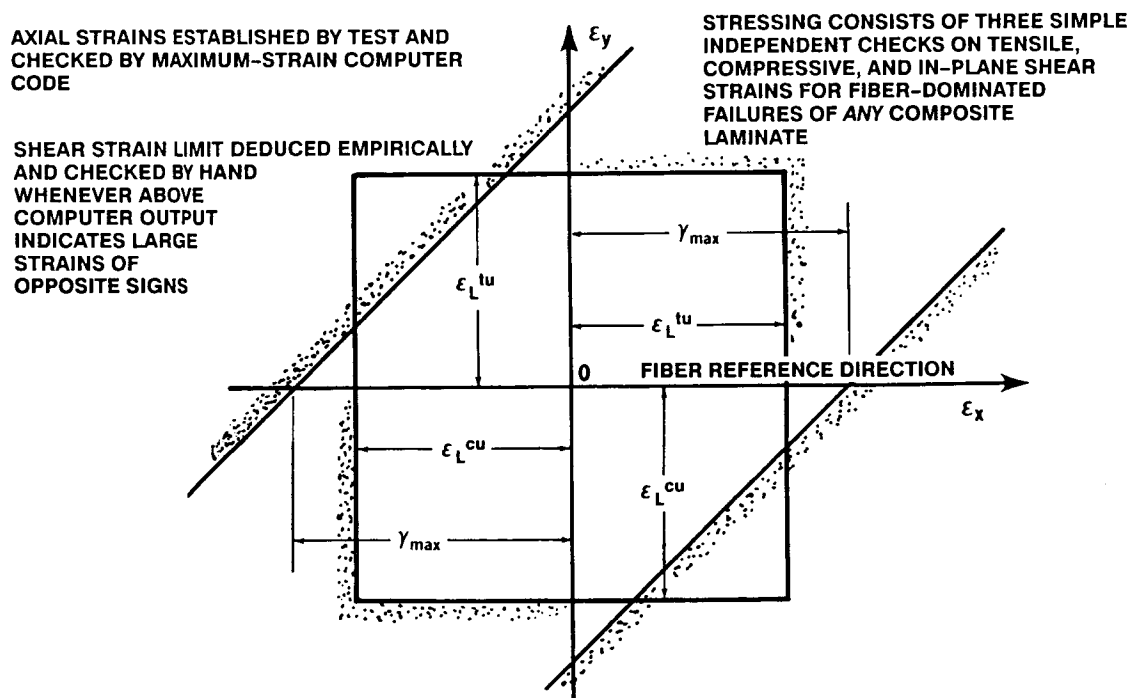


FIGURE 6. BLACK'S FIBROUS COMPOSITE FAILURE CRITERION, AS USED ON THE C-17

## THE TEN-PERCENT RULE

The many abstract mathematical theories surveyed by Rowlands (Reference 4) that purport to be capable of predicting the strength of structural composite laminates would seem to suggest that there is something difficult and mysterious about the task. On the contrary, provided that one does not lose track of physical realism in the model, it is easy to generate plausible sets of uniaxial and biaxial predicted strengths with the simplest of mathematical techniques, as the following approximate methods developed by the author are intended to show. Admittedly, it is necessary to restrict the theory to fiber-dominated failures a priori, but no skilled composite designer would knowingly waste expensive fibers in inferior structures in which the inexpensive matrix really does fail first. This theory is also restricted to fiber-polymer composites in which strong, stiff fibers are embedded in relatively soft matrices. But this limitation also is met by the great majority of composite materials, such as carbon-epoxy, fiberglass-epoxy, and boron-epoxy. There is also the customary restriction to fibers patterns in the  $0^\circ$ ,  $\pm 45^\circ$ , and  $90^\circ$  family, with equal numbers of fibers in the  $+45^\circ$  and  $-45^\circ$  directions. Some of the simplifications are lost for arbitrary laminate patterns, and a computer code is needed to apply the same physical model in such cases.

The basis of the simple analysis is that, for a load in the 0-degree direction, the longitudinal strength and stiffness of cross-plyed laminates can be deduced by applying a simple factor to the appropriate unidirectional 0-degree strength and modulus of a unidirectional tape laminate. The reference strength and stiffness are adjusted for the effects of the environment and must be established experimentally, as for any other composite failure theory. The strengths may need to account for the load direction (tension or compression) as well. Laminates made from biwoven fabrics can be analyzed the same way by first analytically decomposing the cloth into equivalent orthogonal tape layers. This simple theory is set apart from the others by its ease in computing the scaling factor. Each 0-degree ply counts as one unit of strength and stiffness, while every cross-ply\* contributes only 1/10th as much to the strength and stiffness. On that basis, a  $0^\circ/90^\circ$  laminate would have  $(1 + 0.1) / 2 = 0.55$  as much strength and stiffness as an all- $0^\circ$  laminate. Likewise, a quasi-isotropic laminate would be predicted to have  $(1 + 3 \times 0.1) / 4 = 0.325$  as much strength and stiffness as an all- $0^\circ$  laminate. An all- $90^\circ$  or entirely  $\pm 45^\circ$  laminate would be expected to have about 1/10th of the strength and stiffness of the all- $0^\circ$  laminate, but those particular properties are really matrix-dominated and the predictions may not always be relied on for such plies in isolation.

Biaxial strengths for stresses having the same sign are then predicted on the basis of the maximum-strain theory for fibrous composites. For uniaxial loads with a strain  $\epsilon$  in the 0-degree direction, any  $90^\circ$  fibers would be strained by  $-\nu\epsilon$ , while  $\pm 45^\circ$  fibers would develop an axial strain of  $(1 - \nu)\epsilon / 2$  times the strain in the  $0^\circ$  fibers, as explained in Figure 4 of Reference 13. Here, the Poisson's ratio  $\nu$  refers to  $\nu_{xy}$  of the laminate. For a quasi-isotropic laminate,  $\nu_{xy}$  is inevitably very close to 0.33, as derived in Reference 6, while it is only 0.05 for the  $0^\circ/90^\circ$  laminate but almost 0.8 for an entirely  $\pm 45^\circ$  laminate.

Consequently, a uniaxial 0-degree load on a quasi-isotropic laminate would strain the  $\pm 45^\circ$  fibers to only one-third of their ultimate capacity. Since the stress on those fibers would contribute only half as much after rotation to the reference 0-degree direction, an improved estimate of the scaling factor for the strength and stiffness of this laminate would be  $(1.0 + 0.1 + 2 \times 0.333 \times 0.5) / 4 = 0.358$ , a value adequate for design purposes. However, if we examine the biaxial rather than the uniaxial stress state, *all* the fibers must now be stressed equally and the scaling factor would then become  $(1.0 + 0.1 + 2 \times 1 \times 0.5) / 4 = 0.525$ , or some 50 percent *higher* than the strength under a uniaxial load.

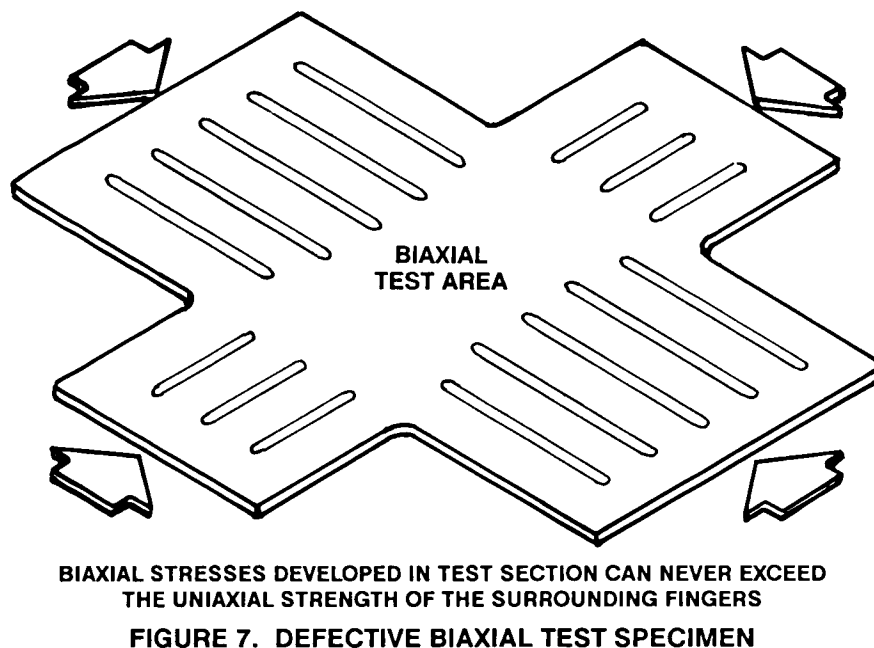
This easily established increase in strength can *never* be demonstrated by the common kind of "biaxial" test specimen shown in Figure 4 of Rowlands' work (see Reference 4), the key features of which are summarized

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\* Here, the author is using the term cross-ply in its general form, to denote *any* ply other than a 0-degree ply. This is contrary to efforts to confine the designation to only 90-degree plies and to use the confusing term "angle ply" to designate all other directions *except* 0 and 90 degrees.

in Figure 7 of this paper. The biaxially loaded interior of the test coupon cannot possibly experience a higher stress than that needed to fail the uniaxially loaded fingers around the periphery. The most obvious demonstration of this deficiency is testing for the biaxial strength of a  $\pm 45^\circ$  laminate which, by definition, must be the same as for a  $0^\circ/90^\circ$  laminate. The surrounding  $\pm 45^\circ$  fingers would have less than one-fifth of the required strength to fail the interior test section of this specimen. Even the quasi-isotropic laminate is 50-percent stronger under equal biaxial stressing than when loaded uniaxially.

This widespread error in trying to experimentally determine the biaxial strength of composite materials using test coupons which are inherently incapable of providing the correct result has been a major reason why so many scientifically unsound composite "failure theories" have not been exposed.

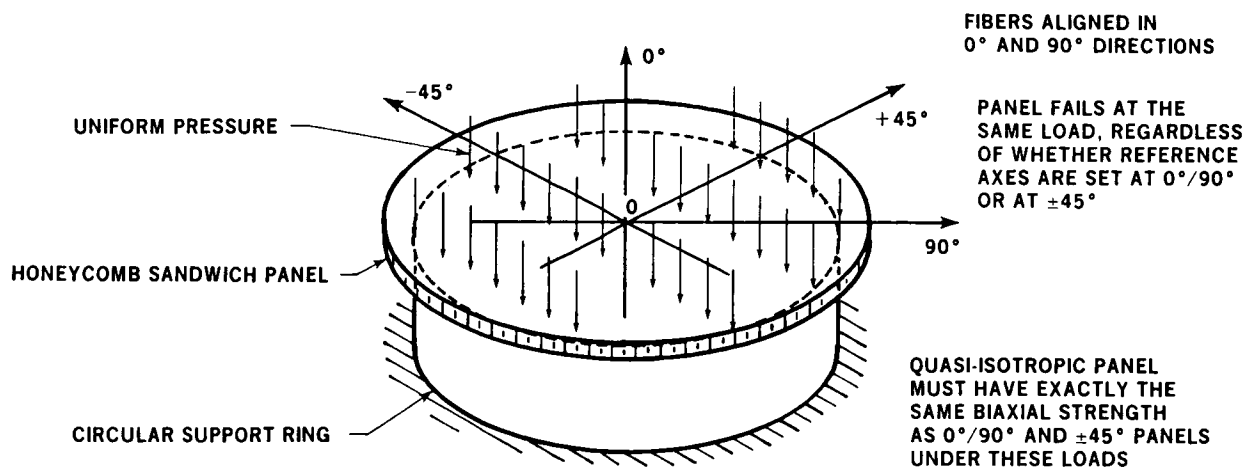


In Reference 14, the author proves that biaxial testing would indeed be a very difficult task, requiring a large circular sandwich plate supported around its periphery and loaded by lateral pressure, as shown in Figure 8, if premature failure at some uniaxially stressed area is not to precede failure in the biaxially loaded central test section. Such an expensive specimen has yet to be tested, although the author is confident that it will eventually be used by those who design and build submarines since knowing the true biaxial compressive strength of composite laminates is so critical to the success of their activities.

However, in Reference 15, Swanson and Nelson used pressurized tubes with varying tensile axial loads to prove beyond any reasonable doubt that the maximum-strain theory is acceptable — and unlikely to be improved upon — for carbon-epoxy composites in the tension-tension quadrant (see Figure 9). Interestingly, their finding that the Tsai-Wu "last-ply" failure model was totally inconsistent with Swanson's test data as long ago as 1986 seems to have had even less effect on the technical community than this author's efforts.

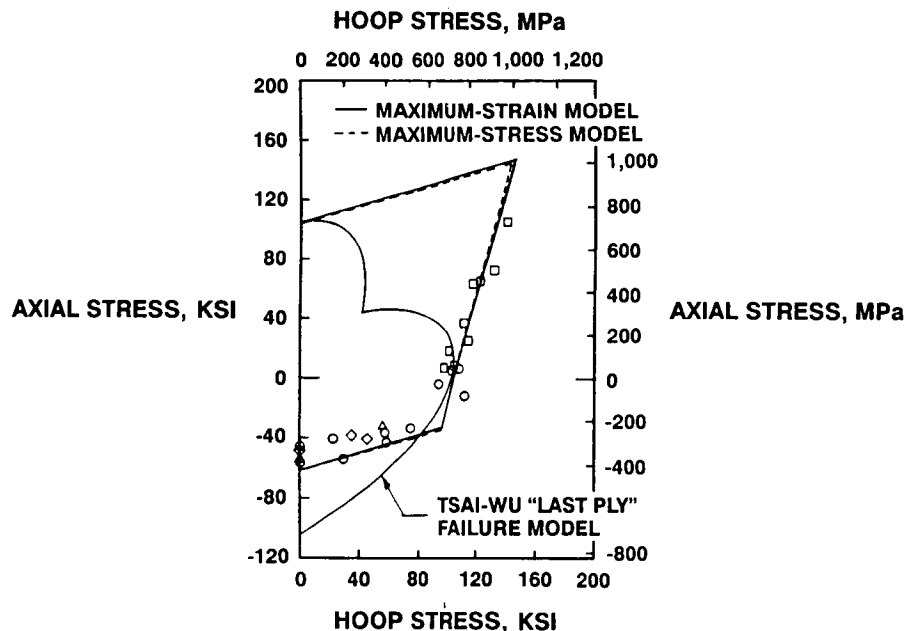
Some test data in the compression-compression quadrant seem to support the author's predictions about the straight-sided form of the failure envelope. However, the strengths measured are all too low, as are Swanson's in this stress domain. New tests are needed to truly characterize composite materials under biaxial compressive stresses.

Nevertheless, the author contends that there is really no need to directly measure the biaxial strengths of fibrous composites since they can be determined with an extremely high confidence level from uniaxial testing of  $0^\circ/90^\circ$  flat laminates. Because the Poisson's ratio is almost zero in this case, the biaxial strength cannot differ



CONSEQUENTLY, ANY PANEL THAT IS A COMBINATION OF 0°/90° AND ±45° LAYERS HAS THE SAME BIAxIAL STRENGTH. THAT STRENGTH IS A LITTLE GREATER THAN THE UNIAXIAL STRENGTH OF A 0°/90° LAMINATE

FIGURE 8. BIAxIAL TEST SPECIMEN DEMONSTRATING IDENTICAL BIAxIAL STRENGTHS OF 0°/90° AND ±45° LAMINATES



SOURCE: SWANSON, OF THE UNIVERSITY OF UTAH

FIGURE 9. SWANSON'S COMPARISON OF LAMINATE FAILURE THEORIES WITH FAILURE STRESSES IN QUASI-ISOTROPIC CYLINDERS

significantly from the uniaxial strength. According to the maximum-strain model, the biaxial strength would be higher than the uniaxial by the ratio  $1 / (1 - \nu)$ , or about 1.05. Now, the biaxial strength of an entirely ±45° laminate must be precisely the same because the only difference is the reference direction for the fiber axes (see Figure 8). Similarly, the biaxial strength of a quasi-isotropic laminate must also be the same, since it must be the average of these two identical quantities.

This biaxial strength serves as a kind of magic number for *all* laminates containing the same number of 0° and 90° fibers, with the remainder shared equally between the +45° and -45° directions. Once the biaxial

strength has been obtained, the uniaxial strength is derived with great precision by multiplying the biaxial strength by  $(1 - \nu)$ , as explained below, there being only one Poisson's ratio for this family of doubly symmetric laminates.

In the case of the  $0^\circ/90^\circ$  laminate, one would expect the normalizing factor with respect to the unidirectional lamina strength to be  $0.55 / (1 - 0.05) = 0.575$ , quite close to the 0.525 deduced earlier. Likewise, the uniaxial strength of the quasi-isotropic laminate would be  $(1 - 0.33) \times 0.575 = 0.383$ , again only slightly above the 0.325 deduced above by treating the Ten Percent Rule in its simplest form as a rule of mixtures and the factor 0.358 deduced by resolving and summing the stresses in the various fiber directions. These various forms of simplified analysis are self-consistent, the result of being based on a physically realistic model.

Turning now to the in-plane shear strength for the same family of doubly symmetric laminates, the author has suggested a strength be selected that is equal to *half* the unidirectional strength of the complementary fiber pattern, with  $0^\circ/90^\circ$  and  $\pm 45^\circ$  fiber contents interchanged (Reference 16). Thus, the fiber-dominated in-plane shear strength of an entirely  $\pm 45^\circ$  laminate would be half the unidirectional tension or compressive strength, whichever is greater, of a  $0^\circ/90^\circ$  laminate. The scaling factor for this laminate, with respect to the uniaxial strength of a unidirectional laminate, would be  $0.5 \times 0.55 = 0.275$ . Similarly, the factor for a quasi-isotropic laminate would be  $0.5 \times 0.325 = 0.163$ , although either of the higher estimates for the second factor would be equally acceptable. The prediction of  $0.5 \times 0.1 = 0.05$  for the in-plane shear strength of an all- $0^\circ/90^\circ$  laminate happens to be nearly correct, but is really suspect because that particular property is obviously matrix- rather than fiber-dominated and so contravenes the original simplifying assumptions.

Failure envelopes for these three fiber patterns, based on the Ten-Percent Rule, are shown in Figure 10. These envelopes are not at all similar to the predictions of Tsai's theories in Figure 11.

This simple procedure, developed by the author for predicting in-plane shear strengths, has been criticized as being unscientific and unworthy of publication. However, when a paper advocating this approach was presented in 1987 (see Reference 16), structural designers had neither reliable test specimens nor credible theoretical methods with which to establish in-plane shear strengths for laminates. What was desperately needed was something at least good enough for preliminary design. And, if this controversial idea inspired others to improve

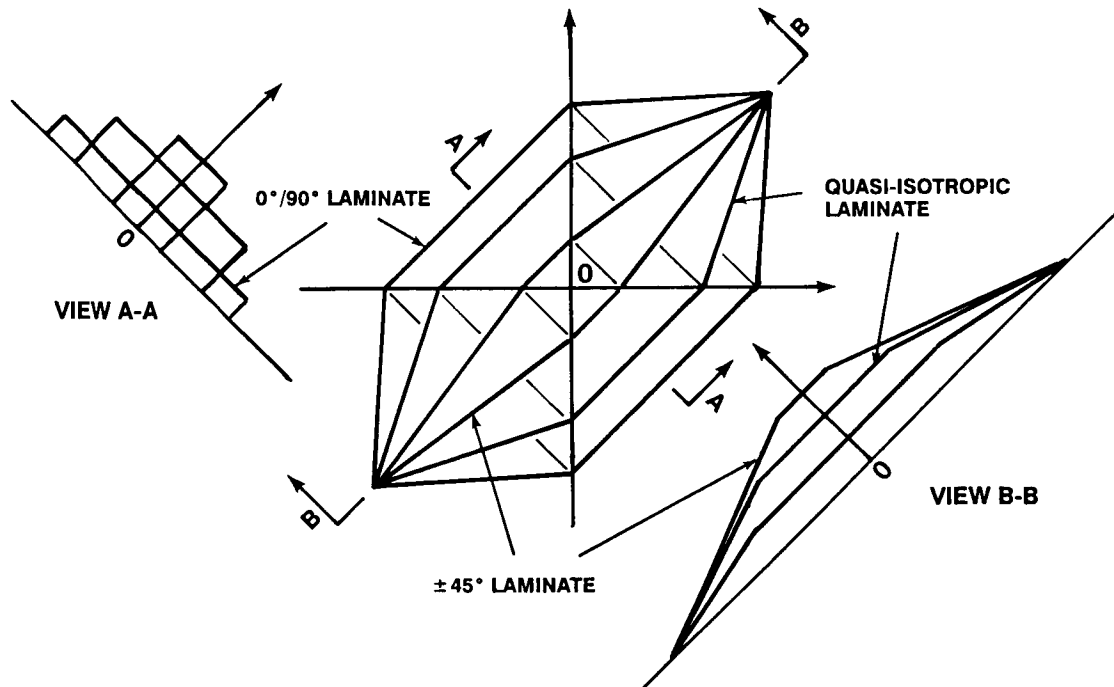


FIGURE 10. FAILURE ENVELOPES ACCORDING TO THE TEN-PERCENT RULE

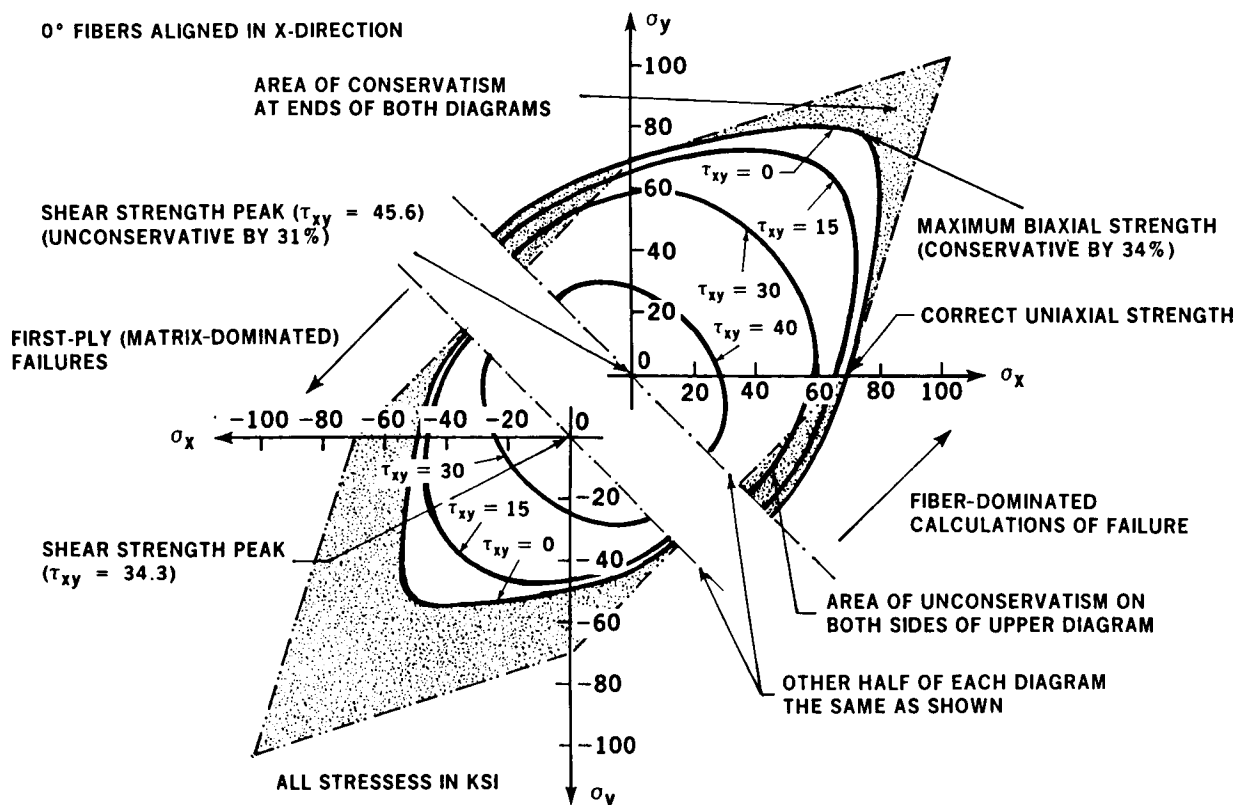


FIGURE 11. FAILURE ENVELOPE FOR QUASI-ISOTROPIC CARBON-EPOXY LAMINATES (TSAI-HILL AND TSAI-WU THEORIES)

both the test specimen designs and the analysis methods, the paper would have served an even greater purpose. Now that such predictions can be made scientifically (see Reference 17), the crude approximations of in-plane shear strength by the Ten-Percent Rule are still close enough to the best analyses to be used for formal stressing. (For a comparison between tests and theories, see Figure 15 of Reference 17).

Even the maximum-strain model of Reference 18, which is outstanding in the tension-tension and compression-compression quadrants, grossly *overestimates* in-plane-shear strengths, typically by 60 percent. Figure 9 of Reference 6 shows that, for the maximum-strain failure model, the in-plane shear strength is  $(1 - \nu) / (1 + \nu)$  times as high as the biaxial strength for *all* doubly symmetric cross-plyed laminate patterns. Therefore, in place of the factors 0.275, 0.163, and 0.05 above, the maximum-strain model would overestimate the in-plane-shear strength via the factors  $[(1 - 0.05) / (1 + 0.05)] \times 0.55 / (1 - 0.05) = 0.524$ ,  $[(1 - 0.33) / (1 + 0.33)] \times 0.55 / (1 - 0.05) = 0.290$ , and  $[(1 - 0.8) / (1 + 0.8)] \times 0.55 / (1 - 0.05) = 0.064$ , respectively.

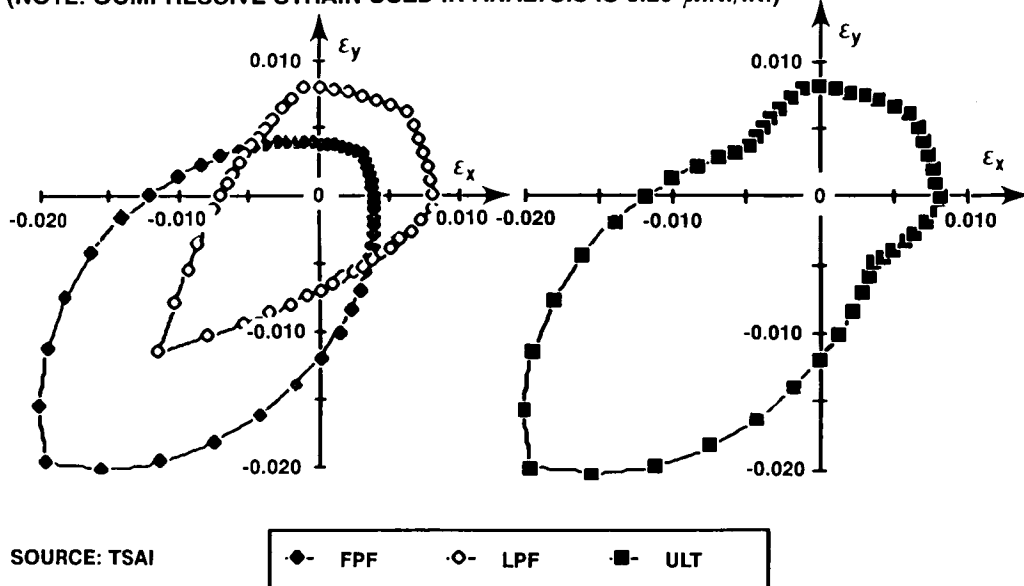
While the sample solutions here are confined to doubly symmetric fiber patterns for simplicity's sake, the original derivations of the simplified analysis methods also cover fiber patterns with different  $0^\circ$  and  $90^\circ$  fiber contents. However, the evaluation of the biaxial strengths for these laminates requires a pocket calculator rather than mental arithmetic.

## COMPARISON WITH OTHER PREDICTIONS OF BIAXIAL COMPOSITE STRENGTHS

This section reveals gross deficiencies in corresponding predictions from a widely promoted computer code based on mathematical theories of anisotropic elasticity for homogeneous materials. Although the illustrative examples refer to only one such computer program, that of Tsai (Reference 19), the criticisms apply equally to all similar codes as well, many of which are cited in Reference 8.

Figure 12 contains first- and last-ply failure predictions published by Tsai (Reference 19) for a quasi-isotropic carbon-epoxy laminate. Tsai advocates accepting the larger estimate for monotonically loaded test coupons on p. 12-6 of his work, as shown on the right of this figure. This recommendation is based on his proposed progressive-failure models. The author's corresponding predictions, using the *same* input properties but ignoring those for which there was no call, are given in Figure 13 for comparison. There are great differences, particularly with respect to the first-ply failure predictions, which do not even permit agreement under uniaxial loads. The

(NOTE: COMPRESSIVE STRAIN USED IN ANALYSIS IS  $8.29 \mu\text{IN./IN.}$ )



(a) FPF AND LPF ENVELOPES OF QUASI-ISOTROPIC T300/N5208 CFRP LAMINATE AND  
(b) THE ULTIMATE STRENGTH ENVELOPE DEFINED BY THE OUTERMOST BOUNDARY OF THE FPF AND LPF ENVELOPES

FIGURE 12. TSAI'S FIRST- AND LAST-PLY FAILURE ANALYSES

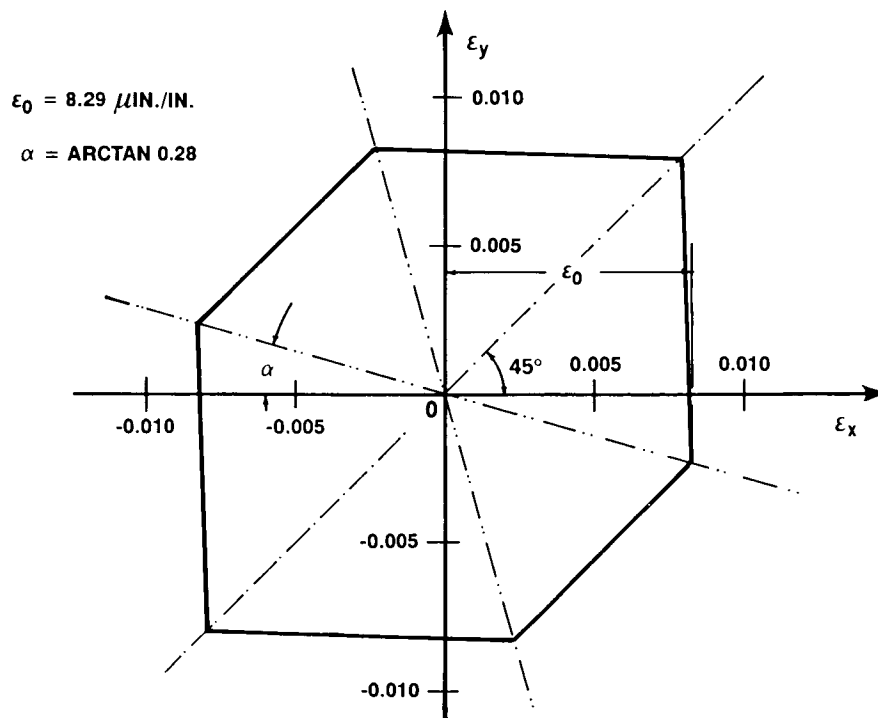


FIGURE 13. HART-SMITH'S FAILURE ENVELOPE FOR A QUASI-ISOTROPIC CARBON-EPOXY LAMINATE

agreement with the last-ply failure predictions is better, but the failure envelope is far from smooth and continuous at the laminate level, which encourages one to question the importance of such a constraint at the unidirectional lamina level.

Tsai's first-ply predictions fall far short of the strengths predicted by the author throughout the tension-tension quadrant and greatly exceed them throughout the compression-compression quadrant. Remarkably, the strains to failure under biaxial compressive loads exceed the input unidirectional compressive strain limit by a factor of nearly 2! No explanation of this is provided and, in the author's opinion, none ever could be. And the justification given by Tsai for reducing these acknowledged excessive estimates *requires* the kind of matrix degradation that could occur *only* during some prior application of *tensile* loads in order to crack the matrix and reduce its ability to support longitudinally compressed fibers in a *subsequent* application of load.

Further, the predicted last-ply biaxial tension strength still falls short of the author's prediction, even when the transverse properties have been adjusted to match the uniaxial tensile strengths reasonably well. Surprisingly, the best agreement seems to be with the in-plane shear (equal and opposite tension and compression) state of stress.

It is difficult to make concrete comparisons with any theory using a progressive failure model because, as indicated in Figure 14, also taken from Tsai's *Composite Design* book, his failure envelope will collapse onto that for the maximum strain theory if one presupposes the necessary amount of matrix degradation. (There seems to be an error in coding his program because the strain in the compression-compression quadrant has collapsed onto the value of strain entered for transverse tension, not longitudinal compression.) However, the agreement achieved in the tension-tension quadrant by invoking matrix degradation has invalidated the agreement previously reached for the in-plane shear loads.

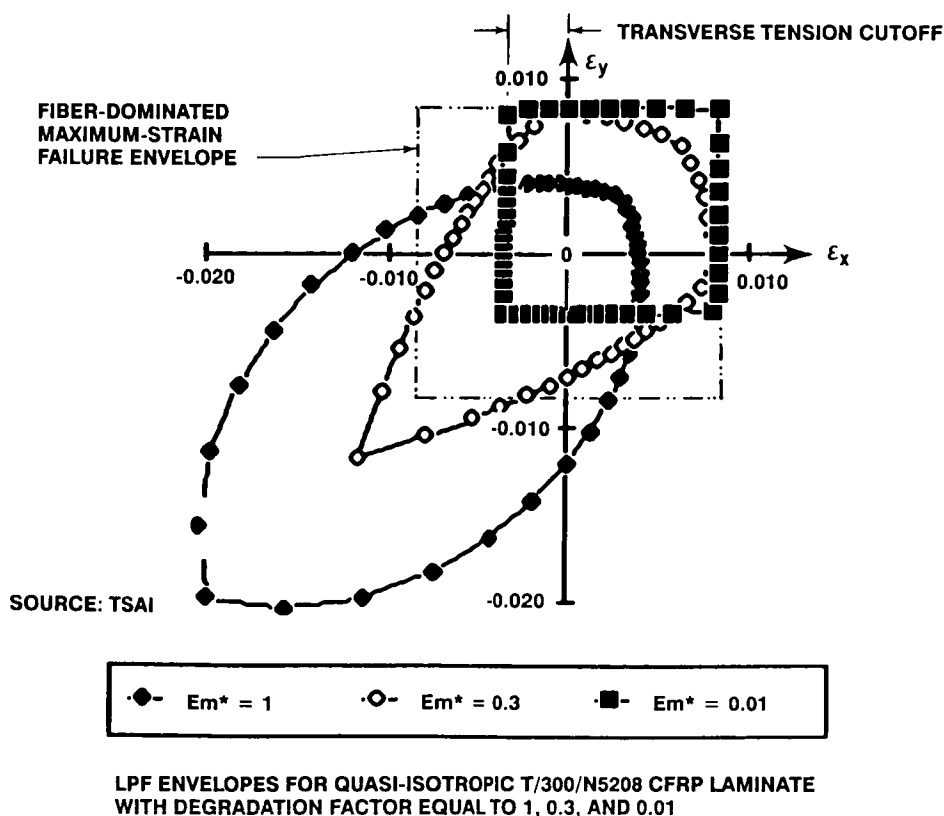


FIGURE 14. TSAI'S FIRST- AND LAST-PLY FAILURE ANALYSES SHOWING PROGRESSIVE TRANSFORMATION INTO MAXIMUM-STRAIN FAILURE ENVELOPE



Given the difficulty of making comparisons with a moving target, as the degree of matrix degradation is altered, one is entitled to ask whether or not it would have been easier to go directly to the maximum-strain failure model instead of arriving at it indirectly via adjustments to the transverse properties entered into some other failure model which would not at first give acceptable answers. The abuse of progressive failure theories to "enhance" predicted composite laminate strengths will be discussed in a future work.

A comparison of Figures 12 through 14 indicates that no matter what degree of matrix degradation is assumed, no *single* set of input properties for Tsai's theory will *simultaneously* match the author's predictions for uniaxial *and* biaxial states of stress. Even when the transverse properties are suitably "adjusted" to match the uniaxial tensile and compressive strengths, the predicted biaxial tension strength will still be too small and the biaxial compressive strength too large.

In theory, one could always add two more adjustable parameters to the lamina failure model, to be set to match the biaxial testing of two cross-ply laminates. Ashizawa, in Reference 20, did so by using a cutoff based on measured fiber-dominated in-plane shear strengths of  $\pm 45^\circ$  laminates. However, while this technique worked whenever the measured shear strength was accurate, the predictions were obviously inconsistent whenever the shear measurement was far too low. Unless one had a valid physical model to guide the process, it is likely that those particular biaxial tests would be inconsistent with predictions based on *other* biaxial tests. And, if one really did have a reliable physical model, one would not need any additional terms.

The quadratic "failure criteria" for fibrous composites are not the first unsound theories which are capable of predicting *some* numerically correct answers to problems despite a consistent inability to solve *other* problems. The beliefs of the Flat Earth Society come readily to mind. While the old idea that the sun revolves around the earth is no longer taught, *many* celestial and seasonal observations were explained at the time by use of this model. The author can only hope that it will take less time to recognize the correct way to predict the strength of composite laminates than it took to reach agreement on a model for the solar system.

Another failing of these abstract mathematical failure "criteria" is exposed by a physical assessment of the most severe "triaxial" stresses that can be applied.\* As the author noted in 1985 in Figure 13 of Reference 13, the cross section of the failure envelope looking along the biaxial stress line *must* be rectangular, as shown in Figure 10. For the case of a quasi-isotropic laminate, the specific load of equal and opposite  $\sigma_x$  and  $\sigma_y$  stresses in the absence of any in-plane shear stress  $\tau_{xy}$  induces *no* load in either the  $+45^\circ$  or  $-45^\circ$  fibers. Most of it is reacted by axial tension in the  $0^\circ$  fibers, for example, and simultaneous compression in the  $90^\circ$  fibers. A very small fraction of the load is reacted by shearing the resin matrix. On the other hand, the application of a pure in-plane shear load to the same laminate, with respect to the same reference axes, would load up the  $\pm 45^\circ$  fibers while leaving the  $0^\circ$  and  $90^\circ$  fibers unloaded. This particular cross section of the failure envelope is therefore rectangular because there is essentially *no* interaction between the loads, one of which is carried by the  $0^\circ/90^\circ$  fibers while the other is resisted by the  $\pm 45^\circ$  fibers. The form of the abstract mathematical failure envelopes in Figure 11 is in stark contrast to Figure 10, being smoothly curved all over.

The failure envelopes in Figure 11 are obviously also in error at the biaxial tension and biaxial compression points. Since *all* fibers are equally strained under those conditions, it is physically impossible to add in-plane shear loads without decreasing the in-plane direct loads. The ends of the failure envelope must be *pointed*, not rounded as they are in Figure 11.

The reason for these errors is that Tsai's fictitious failure criterion, cited on p. 11-5 of Reference 19, contains a mixture of unrelated reference strengths, some pertaining to the fiber and others to the matrix. Tsai's formula and its many clones are restricted in validity to truly homogeneous materials exhibiting only *one* failure mechanism for *all* the states of combined stresses being considered. And, under such circumstances, the use of four

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\* Actually, it is the combination of biaxial in-plane direct loads with additional in-plane shear, so it is really a case of biaxial loads with respect to different axes.

or five test measurements to characterize the strength of the material should not be necessary. For distinctly heterogeneous materials such as fiber-polymer composites, on the other hand, it is necessary to write *separate* failure criteria against the fibers, the matrix, and possibly also the interface between the two. Further, additional criteria are needed whenever multiple failure modes are possible for any constituent.

There should perhaps be additional interlaminar criteria for the immediate proximity of any boundaries, but this refinement is customarily ignored, along with stacking-sequence effects, which is why every so often composite laminates with excessive clustering of parallel fiber layers delaminate during cooldown before they are even removed from the autoclave. And more often than not, a laminate was designed that way because some computer "optimization" program was used to identify the most "suitable" laminate instead of allowing accumulated experience to dictate that there be a minimum percentage of plies in each of the four standard directions and that there be strict limits on the clustering of parallel plies. Tsai's position on this matter is stated on p. 7-1 of Reference 19. "For symmetric laminates subjected to in-plane loads only, the stacking sequence of plies is not important." He accounts for stacking sequence only when studying the bending of laminates and all but ignores the issue of edge delaminations, while concentrating on intraply failures.

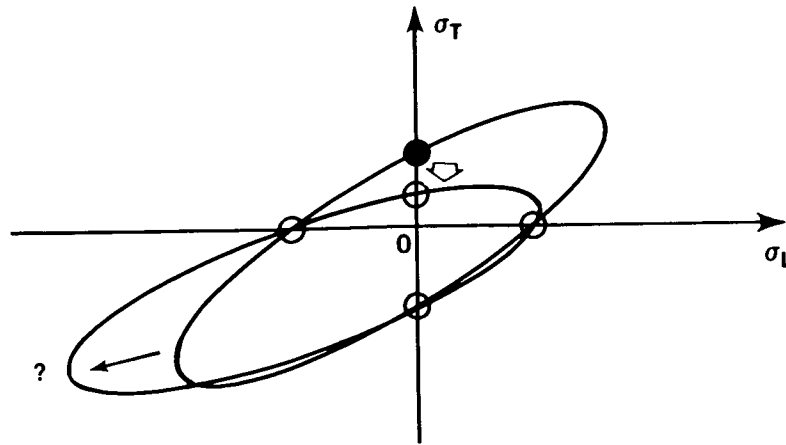
A major difference between the author's methods of predicting the strength of fibrous composites and those typified by the works of Tsai is that, in the author's case, minor changes in "lamina" properties do not affect the basic *form* of the failure envelopes. The failure envelope has remained characteristically flat-faceted since the very first report on the subject in 1984 (see Reference 21). Each facet has been defined by the failure of one particular fiber direction under a uniform failure mode throughout. The intersections of the facets denote the simultaneous failure of two or three fiber directions, depending on how many facets intersect. Reference 22 even includes parametric studies showing the small effects of systematic variations in material properties. Yet a study of Reference 19, for example, will show an endless variety of shapes, some associated with different fiber patterns but many caused merely by a change in the level of "degradation" due to matrix "cracking." Such variability does not inspire confidence in a theory.

The author's failure model for composite laminate analysis has been criticized as being too simplistic by some well-regarded researchers. A particular stumbling block is that the final failure envelope shows only *one* line for compressive failures while there is considerable evidence that several failure modes are possible. This is not germane to the level of analysis being performed here. Also, the critics seem to have missed the very clear coverage of both shear failures of the fibers and some not necessarily defined form of compressive instability, depending on the particular composite material under investigation. The author is interested in only the *weakest* of the possible failure mechanisms, whatever it may be. It might even change with operating environment and certainly changes between unidirectional and cross-ply laminate patterns. There is no need for the author to address this issue of multiple possibilities for any one facet of the failure envelope because the experimentally derived input data will automatically identify the weakest mechanism provided that the test coupon and fixture are representative of the real structure.

The reluctance to admit that the art of predicting the strengths of composite laminates has not been perfected is apparent in a response to the author's attempts to find interest in improving the analytical techniques. If one were to believe the predictions of the Tsai-Wu "failure theory," for example, one would be forced to conclude that the underwater compressive strength of a composite submarine hull could be increased substantially by *reducing* the interlaminar (and hence transverse-tension) strength of the lamina, as explained in Figure 15. While many got the message, one response was; "That sounds like a great idea. How do we reduce the interlaminar tension strength?"

## A MAJOR INCONSISTENCY IN STANDARD COMPOSITE FAILURE THEORIES

Analytical predictions of strength based on a combination of orthogonal unidirectional tape layers have no resemblance whatever to the predicted strengths of the same fibers in the same resin, in the form of a cloth



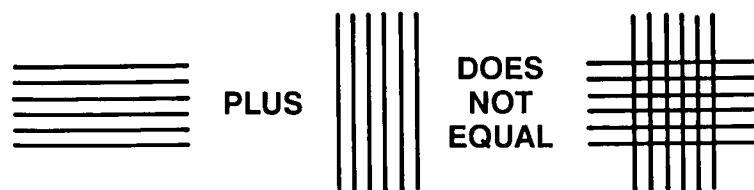
WOULD ANYONE BELIEVING THIS ANALYSIS BE WILLING TO PUT TO SEA FOR THE DIVING TRIALS?

**FIGURE 15. "IMPROVED" COMPOSITE MATERIAL FOR SUBMARINE HULLS BY DECREASING TRANSVERSE-TENSION STRENGTH OF UNIDIRECTIONAL LAMINA**

laminate, even when the two laminates have precisely the same elastic constants and the fibers have precisely the same failure stress or strain. The issue has nothing to do with kinks in the woven fabric; the dissimilarity also exists when the bidirectional lamina is made from uninked dry stitched preforms which are subsequently impregnated with resin.

Figure 16 illustrates this point very clearly. The irreconcilability is quantified in Figure 17 by various analyses of 0° and 90° and 0°/90° laminates. The analyses are symmetric about the diagonal running from the lower left to the upper right, so only half of each is shown, with the "tape" analyses in the lower right and the equivalent "cloth" analyses in the upper left. Precisely the same fiber strains-to-failure are used throughout the analyses, and the output of the tape laminates analysis is used to define the elastic constant inputs for the cloth analysis. The first-ply failure (FPF) analyses on the right of the figure show a gross underestimate of the tensile strengths,\* with respect to both the author's theory and the well-known maximum-strain failure model, which is compensated for by a gross overestimate of the compressive strengths. The computer code then "modifies" the tape material properties and recalculates last-ply failures (LPF) which appear to agree much better with the author's theory. Tsai's reduction in transverse stiffness to achieve this transformation is directly equivalent to the author's recommendations in Reference 22. There was concern that the predictions of the BLACKART computer code would be invalidated by premature transverse-tension failures of the type responsible for the distortion of the FPF envelope in Figure 17. But the author's approach has been described as merely a fudge. Perhaps it would have appeared more scientific if it had been accomplished by a computer code.

The FPF predictions shown in the upper left of Figure 17 should correspond to fiber-dominated failures and, indeed, they agree well with the author's predictions, apart from the minor problem of physically impossible small overestimates of strength whenever the two in-plane stress components have the same rather than opposite



**FIGURE 16. FATAL FLAW IN TENSOR-POLYNOMIAL COMPOSITE FAILURE CRITERIA**

\* The cusp at the biaxial tension point in Figure 9 results from Swanson's interpretation that if all the laminae are *equally* critical according to the FPF analysis, no enhancement of strength is possible under an LPF analysis. Only the transverse ply properties were degraded in Figure 17.

(BOTH ANALYSES SHOULD HAVE BEEN THE SAME IF THEY HAD SCIENTIFIC VALIDITY)

HART-SMITH'S TRUNCATED MAXIMUM-STRAIN MODEL

SINGLE ORTHOTROPIC MATERIAL HAVING THE SAME ELASTIC PROPERTIES AND FIBER STRAINS-TO-FAILURE AS THE CROSS-PLYED TAPE LAMINATE



LINE OF SYMMETRY

PHYSICALLY IMPOSSIBLE REGIONS

$\epsilon_y$

TWO ORTHOGONAL LAYERS OF UNIDIRECTIONAL TAPE

TENSOR-POLYNOMIAL "ANALYSES" USING TSAI-WU FAILURE MODEL  
(MATRIX DEGRADATION FACTOR = 0.2)



PLUS

$\epsilon_x$

FIGURE 17. CONFLICTING ANALYSES OF THE SAME 0°/90° COMPOSITE LAMINATE

signs. However, this apparent reconciliation is undermined by further computer coding in the form of LPF predictions for the cloth laminate which are significantly weaker. No justification for this second analysis has been found in Tsai's book, and no physical explanation is given in the text accompanying the computer code.

However, the good agreement between the "cloth" FPF analysis in Figure 17 and the author's analysis reinforces the author's repeated claims over the years that the mechanical properties needed for predicting the strength of cross-plyed composite laminates are those prevailing in the cross-plyed laminate, not those for the unidirectional tape laminate in isolation. If, instead of proceeding from the tape to a cloth analysis, the process had been reversed to establish representative in-situ mechanical properties for the unidirectional tape *in the presence of orthogonal fibers*, the "tape" analysis for the combination of unidirectional 0° and 90° plies would also have been correct *because all the predicted failures would have been fiber-dominated*. The only thing found wrong would be the original hypothesis that it was appropriate to create a composite failure model based on measured tensile and compressive strengths in the longitudinal and transverse directions for a unidirectional tape laminate.

## CONCLUSIONS

One would have to conclude from the simplicity of the author's Ten-Percent Rule for approximate analysis of fibrous composite laminates that it should be extremely difficult to develop computerized composite "failure theories" that are incapable of correctly predicting the uniaxial strengths under tension and compression. Surprisingly, the literature on the subject shows that many authors have failed to predict even these simplest of laminate strengths.

Of the well-established failure models, only the empirical maximum-strain model has been found by the author and others to lead to acceptable predictions for even the simplest load cases. This same theory has

already been confirmed by experiment to be valid throughout the tension-tension quadrant. And the author expects that it will eventually be proven equally valid throughout the compression-compression quadrant when reliable test data are generated. However, this theory has been shown to be unacceptably unconservative, typically by 60 percent, for the in-plane-shear loads in which the biaxial stresses are of opposite sign.

If it were not for this in-plane-shear case, there would be no criterion by which to distinguish between the maximum-strain theory and the author's generalization of the Tresca (maximum-shear-stress) failure model. This particular biaxial stress state is *crucial* in selecting one of the two theories. The criticality applies equally in assessing other physically plausible failure models, some of which have considerable experimental support for *other* states of stress. For instance, as shown in Figure 5, a failure model based on a combination of notch sensitivity in the tension-tension quadrant and some form of compressive instability throughout the compression-compression domain cannot be challenged except for the need to find a third mechanism to eliminate unacceptably unconservative predictions throughout the tension-compression (shear) states of biaxial stress. Neither the author nor anyone else is in a position to challenge such theories if they are not assessed for *all* states of uniaxial and biaxial stresses.

In applying and presenting the generalized maximum-shear-stress failure model, the author has already acknowledged the need to provide for at least one *different* failure mechanism, that of compressive instability for the newer high-strain carbon fibers of very small diameters. Expressing his new failure theory with respect to the strain plane, even though it is really a stress-based criterion, makes it easy to superimpose additional realistic failure modes. The author also acknowledges the need to provide for but only rarely use transverse matrix cracking for such composite materials as S-glass fibers in a brittle polymer matrix.

The author's failure model is set apart from the numerous abstract mathematical theories of anisotropic elasticity by additional failure modes that can be superimposed on the basic shear failure envelope for the fibers *without* altering the predicted strengths for other failure modes. With the fictitious smooth curves drawn through four unrelated measured strengths, on the other hand, a change in any *one* of these four calibration points alters *every* predicted strength except for the other three reference strengths. A weakness in transverse tension strength, for example, would be expected to cause predicted increases in the biaxial compression strength, while premature compressive failure by instability would be associated with a prediction that the biaxial tension strength had been enhanced.

It should now be evident that the innumerable abstract mathematical "failure theories" for fibrous composites, which have been acknowledged as unreliable by their own originators the moment they had to invoke "progressive failure" models to achieve agreement with even the uniaxial tests, are beyond redemption as useful structural design tools when one also considers the biaxial stress states.

## ACKNOWLEDGMENT

The author is indebted to J. B. Black, Jr., a kindred spirit whose high regard for publications on this subject is indicated by his Douglas Aircraft Company report entitled "Failure Criterion No. 1,075,372." His understanding of composite failure mechanisms permitted the rational analysis of the composite components on the C-17 aircraft, using an empirically established truncated maximum-strain model of the type shown in Figure 6, *before* the author had reexpressed his own generalized maximum-shear-stress failure criterion in the strain rather than stress plane.

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## STATIC AND DYNAMIC STRAIN ENERGY RELEASE RATES IN TOUGHENED THERMOSETTING COMPOSITE LAMINATES

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### Abstract

*In this work, the static and dynamic fracture properties of several thermosetting resin based composite laminates are presented. Two classes of materials are explored. These are homogeneous, thermosetting resins and toughened, multi-phase, thermosetting resin systems. Multi-phase resin materials have shown enhancement over homogenous materials with respect to damage resistance. The development of new dynamic tests are presented for composite laminates based on Width Tapered Double Cantilevered Beam (WTDCB) for Mode I fracture and the End Notched Flexure (ENF) specimen. The WTDCB sample was loaded via a low inertia, pneumatic cylinder to produce rapid cross-head displacements. A high rate, piezo-electric load cell and an accelerometer were mounted on the specimen. A digital oscilloscope was used for data acquisition. Typical static and dynamic load versus displacement plots are presented. The ENF specimen was impacted in three point bending with an instrumented impact tower. Fracture initiation and propagation energies under static and dynamic conditions were determined analytically and experimentally. The test results for Mode I fracture are relatively insensitive to strain rate effects for the laminates tested in this study. The test results from Mode II fracture indicate that the toughened systems provide superior fracture initiation and higher resistance to propagation under dynamic conditions. While the static fracture properties of the homogeneous systems may be relatively high, the apparent Mode II dynamic critical strain energy release rate drops significantly. The results indicate that static Mode II fracture testing is inadequate for determining the fracture performance of composite structures subjected to conditions such as low velocity impact. A good correlation between the basic Mode II dynamic fracture properties and the performance in a combined material/structural Compression After Impact (CAI) test is found. These results underscore the importance of examining rate-dependent behavior for determining the longevity of structures manufactured from composite materials.*

### Introduction

With composite materials being used in primary aerospace structures, some basic understanding of fracture is necessary. This is especially important to develop methodologies for determining damage resistance and damage tolerance of composite structures [1]. "Damage resistance" refers to the ability of a material/structure to sustain an "event" without resulting in damage and "damage tolerance" refers to the ability of a material/structure to maintain performance with damage present. Composite laminates typically have poor, through-the-thickness performance. This makes them especially sensitive to out-of-plane loadings such as bending and impact. The goal is to provide an understanding of fracture such that the performance of laminated structures subjected to low velocity impact can be obtained. For a preliminary assessment of performance, a study of Mode I and Mode II fracture behavior under static and dynamic conditions was conducted. These modes of fracture are important for delamination initiation and propagation for thin, composite laminates subjected to low velocity impact [1-5].

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### Toughened Thermosetting Resin Systems

A relatively new class of thermosetting resin systems for advanced composites is being developed [2-4]. These toughened systems are being developed with better damage resistance and damage tolerance to improve the longevity of aerospace structures such as commercial aircraft. Bradley has noted that an improved process zone in composite laminates is a key for determining interlaminar fracture performance [5]. This process zone is the region between plies and is where interlaminar fracture (delamination) typically occurs. Figure 1a is a photograph of a crack (delamination) in the interply of Hercules IM7/3501-6. Notice that the crack propagates close to the fibers, along one side of the fiber/matrix in a self-similar manner. Some separation of the fracture surface is seen, but the fracture propagates at the ply/matrix interface. This is typical of brittle, homogenous, thermosetting resin systems [6]. Figure 1b is a photograph of an interply crack in Hercules IM7/8551-7. Notice how the interply crack is not self-similar and propagates through and around the different phases. This combination of tough phases and process zone enhancement results in laminates which have superior damage resistance during low velocity impact [7]. Consequently, a basic study to determine the fracture initiation and propagation in this interply region under static and dynamic conditions is necessary for a preliminary understanding of the mechanisms of toughening.

### Mode I Test Method

For Mode I fracture performance, the Width Tapered Double Cantilevered Beam Test (WTDCB) was used [8]. A schematic of the geometry used in this study is shown in Figure 2. This sample exhibits a constant fracture load for a given Mode I critical strain energy release ( $G_{Ic}$ ). From elementary beam theory, for the geometry given in Figure 2, the Mode I critical strain energy release rate,  $G_{Ic}$ , is calculated to be:

$$G_{Ic} = \frac{12P^2}{Eh^3} \left( \frac{a}{b} \right)^2$$

*If the thicknesses of the halves are unequal;*

$$G_{Ic} = \frac{6P^2}{E} \left( \frac{1}{h_1^3} + \frac{1}{h_2^3} \right) \left( \frac{a}{b} \right)^2$$

1)

where  $h$  is the half-thickness of the sample and  $h_1, h_2$  are the thicknesses for unequal thicknesses. Hercules uses a nominal specimen which is 152 mm long, 25.4 mm wide, and 3.3 mm thick with a 25 mm precrack. The taper ratio ( $a/b$ ) is equal to 4.

For static testing, the specimen is simply loaded into a test machine and tested at 2.5 mm/ min. The average load during fracture is recorded and used in Equation 1. A schematic for the Mode I dynamic testing is shown in Figure 3. In this test setup, a low inertia, pneumatic cylinder is used for actuation. The specimen is instrumented with a high-speed, piezo-electric load cell. In addition, an accelerometer is placed on the cross-head to monitor accelerations which are numerically-integrated to determine velocities and displacements. A typical load-time plot is shown in Figure 4. The total fracture time is approximately 9 ms. Notice that, while some deviation in the load is seen during fracture, the load remains relatively constant. Consequently, Equation 1 was used for determining Mode I dynamic critical strain energy release rates ( $G_{Ic}$ ). Some typical results, comparing static and dynamic Mode I dynamic critical strain energy release rates, are shown in Table 1. Five coupons from each specimen type were used for the tests.

**Table 1. Comparison of Static and Dynamic Mode I Fracture Properties**

| Material Type | Static $G_{Ic}$<br>J/m <sup>2</sup> (in-lbs/in <sup>2</sup> ) |        | Dynamic $G_{Ic}$ (Eq.1)<br>J/m <sup>2</sup> (in-lbs/in <sup>2</sup> ) |        |
|---------------|---------------------------------------------------------------|--------|-----------------------------------------------------------------------|--------|
| IM7/X1        | 394                                                           | (2.25) | 312                                                                   | (1.78) |
| IM7/X2        | 256                                                           | (1.46) | 286                                                                   | (1.63) |
| IM7/X4        | 530                                                           | (3.02) | 500                                                                   | (2.85) |
| IM7/8551-7A   | 249                                                           | (1.42) | 202                                                                   | (1.15) |
| IM7/8552      | 233                                                           | (1.33) | 240                                                                   | (1.37) |
| IM7/8551-7    | 552                                                           | (3.15) | 607                                                                   | (3.46) |
| IM7/8551-7C   | 381                                                           | (2.17) | 456                                                                   | (2.60) |

In Table 1, the 8552 is representative of a homogeneous resin system and the X1, X2, X4 and 8551 type resins are representatives of multiphase systems. The coefficient of variation within a panel is typically 5%-6% and the coefficient of variation from panel to panel is typically 10%. With consideration to the variability in this test, the data generally show little sensitivity for the rates tested in this study.

#### Mode II Test Method

For these basic studies, the End Notched Flexure (ENF) specimen was chosen as a result of its ease to manufacture and test [9-15]. This sample is illustrated in Figure 5. In Figure 5, a beam with a crack in one end is loaded in three point bending to produce interlaminar shear at the midplane. The crack length is denoted as  $a$  and the specimen supported length is  $2L$  with an overall thickness of  $2h$ . Under loading, the crack propagates and simple beam theory is used to determine the critical strain energy release rate at fracture. From Linear Elastic Fracture Mechanics (LEFM) and simple beam analysis of the ENF, the Mode II critical strain energy release rate,  $G_{IIc}$  may be calculated as [9-11]:

$$G_{IIc} = \frac{9P^2Ca^2}{2w(2L^3 + 3a^3)} \quad 2)$$

where  $P$  is the load at fracture, and  $C$  is the overall compliance of the specimen. An additional, small correction to account for shearing deformation may be used by replacing  $C$  by  $C^*$  [11]:

$$C^* = C + \frac{1.2L+0.9a}{4whG_{13}} \quad 3)$$

where  $G_{13}$  is the interlaminar shearing stiffness of the specimen. It is noted that, for practical specimen geometries, this correction is less than 2% to the compliance from simple beam theory.

An alternative method for determining the average strain energy release rate during crack propagation may be developed by taking data from the load-displacement curve during testing [16]. In this method,  $P_i$  is the load at onset of fracture, and  $P_a$  is the load at which the fracture is arrested. By assuming a straight line between  $P_i$  and  $P_a$  the strain energy release rate may be determined as:

$$G_{IIc} = \frac{P_i\Delta_a - P_a\Delta_i}{2w(a_a - a_i)} \quad 4)$$

where  $\Delta_i, \Delta_a$  are the measured displacements at initiation and arrest, and  $a_i, a_a$  are the crack lengths at initiation and arrest. Whitney, et. al. [17] have shown this method to be approximately 10% higher than the compliance method as in Equation 2 for Mode I fracture and Maikuma, et. al. [15] have shown the area method based on energies (which are determined from load versus time data) to be approximately 6% lower for dynamic testing in Mode II fracture. These results are well within the scatter of such fracture tests [11, 17].

To determine the critical strain energy release rates under dynamic conditions, a Dynatup ETI 500 instrumented impact tower was used. A 5.3 kg mass with an impact velocity of 1.93 m/sec was used for the load introduction (kinetic energy equal to 9.8 joules). The test specimen geometry used for this study was 149.2 mm long (2L) by 25.4 mm wide (w) by 3.3 mm thick (h) with a notch length of 25.4 mm (a). A thin, 0.025 mm Teflon™ film was used for the insert and the crack was propagated to 25.4 mm using Mode I propagation. Fiber bridging was not seen at the crack tip in the composite systems tested in this study. Hercules IM7 fiber was used for most of the studies. This fiber yields an average laminate flexural stiffness of 152 GPa. The average laminate interlaminar shear stiffness is 5.7 GPa.

A typical force versus time and displacement versus time is shown in Figure 6. Knowing the mass of the impactor and the initial velocity, the displacement of the tup is calculated via numerical integration (trapezoidal rule twice) of the directly measured force versus time data. In this figure, the load oscillates up to a peak and drops sharply at the point of fracture. The fracture was rather catastrophic. Under impact, either the sample did not fracture or the crack propagated in an uncontrolled manner in approximately 1 msec with very little change in displacement. Unlike constant crosshead displacement loading conditions, the fracture propagated over the majority of the length of the sample, beneath the loading tup. This is also illustrated in Figure 7 in the load versus displacement plot. While the load oscillates considerably at fracture, the displacement remains relatively constant. Equation 2 (with Equation 3) may be used to predict the initiation strain energy release rate. Note in Figures 6 and 7, a discontinuity in the behavior is seen during fracture. This discontinuity is a result of the flexural wave propagating beneath the load cell. As the crack propagates beneath the impactor, the shape of the curve does not change. That is, if frictional effects are significant as the crack traverses beneath the loading nose, a significant change in the slope in the subsequent behavior would be expected to be seen in the load - displacement curve (Figure 7) as a result of coulomb dissipation. Consequently, frictional effects as the crack propagates beneath the loading tup were assumed to be no more severe than static testing. However, additional studies may be warranted based on some of the unexpected results presented below.

Using Bernoulli-Euler beam theory, the fractured specimen compliance can be determined as:

$$C = \frac{-16L^3 + 36L^2a - 18La^2 + 3a^3}{8Ewh^3} \quad 5)$$

In Equation 5, the crack is assumed to traverse beneath the impactor. This equation is used in a combined analytical-experimental approach to determine the load at crack arrest. By measuring the crack length and using Equation 5, the arrest load is predicted and plotted in Figures 6 and 7, denoted as  $P_a$ . This corresponds reasonably well with the average dynamic load at fracture arrest. Consequently, the work of fracture,  $W$ , to utilize for average fracture energies during dynamic propagation is:

$$W = \int_0^{\Delta_a} P d\Delta - \frac{P_a}{2} \Delta_a \quad 6)$$

where the integral is the area under the experimental force versus displacement curve and  $P_a$  is determined via Equation 5. Some vibratory behavior is present in Figure 6, and Equation 6 represents the average dynamic strain energy release from initiation to arrest. This Equation was used in lieu of Equation 4. In utilizing this equation, it is implied that dynamic effects in the calculation are negligible. This was shown to be true for dynamic tests under similar conditions to the present work by Maikuma, et. al. [15].

### Experimental Results

Table 2 is a comparison of Hercules composite systems subjected to static and dynamic Mode II fracture. Five (5) coupons were used for each type of test unless noted. The dynamic fracture behavior presented in Figures 6 and 7 is a combined material and structural test and is dependent on specimen geometry, support conditions, impactor metrics, etc. However, all coupons were tested under the same conditions for consistency.

All laminates are nominally 34% resin content by weight (nominally 58.8% fiber volume). The X8553, 8551-7 and X series laminates are multi-phase, process zone enhanced laminates similar to Figure 1b; while the 3501-6 and 8552 resin type laminates are homogeneous with limited process zones in the interply region, similar to Figure 1a [4,5]. The process zone enhanced laminates exhibit higher Mode II toughnesses than the homogeneous systems. The X2 resin is a toughened system formulated to have superior hygrothermal properties (hot/wet 0° compression performance) but exhibits lower Mode II toughness than other systems. Two resin contents were tested in the X2 formulation as noted. The higher resin content X2 system exhibits higher Mode II fracture properties.

In Table 2, there is not necessarily a correlation between static Mode II and dynamic Mode II fracture properties. This is especially evident in the case of the homogeneous systems. While the 8552 exhibits the poorest static Mode II interlaminar fracture toughness, it has better Mode II dynamic properties. Its damage resistance in low velocity impact is superior as well [18]. The AS4/3501-6 exhibits the poorest damage resistance in low velocity impact. Notice that it also has the lowest initial dynamic strain energy release rate as defined by Equation 2 (with the minor shearing correction suggested in Equation 3). Upon initiation, the average fracture resistance as determined by Equations 4 and 6 drops dramatically compared to the other materials. The Average Dynamic  $G_{IIc}$  (Eq. 6) of the AS4/3501-6 is less than half of the Dynamic  $G_{IIc}$  (Eqs. 2,3) and considerably lower than the static Mode II fracture properties. Once fracture initiates, the resistance to propagation is lower in the AS4/3501-6 system, resulting in more damage (delamination) under dynamic conditions [2]. Similar trends have been noted by other investigators [13,15] for the same 3501-6 resin system. The homogeneous resin laminates appear to be much more strain rate sensitive than the toughened systems. Hence, static fracture properties may not be applicable for determining the damage resistance, damage tolerance and longevity under dynamic conditions.

**Table 2. Comparison of Static and Dynamic Mode II Fracture Properties**

| Material Type       | Static $G_{IIc}$<br>J/m <sup>2</sup> (in-lbs/in <sup>2</sup> ) [CV <sup>1</sup> ] |        |       | Dynamic $G_{IIc}$ (Eq. 2)<br>J/m <sup>2</sup> (in-lbs/in <sup>2</sup> ) [CV <sup>1</sup> ] |        |                 | Average Dynamic $G_{IIc}$ (Eq. 6)<br>J/m <sup>2</sup> (in-lbs/in <sup>2</sup> ) [CV <sup>1</sup> ] |        |                 |
|---------------------|-----------------------------------------------------------------------------------|--------|-------|--------------------------------------------------------------------------------------------|--------|-----------------|----------------------------------------------------------------------------------------------------|--------|-----------------|
| IM7/8551-7          | 1867                                                                              | (10.6) | [5.6] | 2730                                                                                       | (15.5) | [3.9]           | 2272                                                                                               | (12.9) | [8.3]           |
| IM9/8551-7          | 1691                                                                              | (9.6)  | [7.5] | 2184                                                                                       | (12.4) | [7.2]           | 1832                                                                                               | (10.4) | [10.5]          |
| IM7/X8551-7C        | 1374                                                                              | (7.8)  | [2.6] | 2360                                                                                       | (13.4) | NA <sup>4</sup> | 1744                                                                                               | (10.2) | [12.7]          |
| IM7/X1              | 1268                                                                              | (7.2)  | [5.9] | 2202                                                                                       | (12.5) | [4.7]           | 2184                                                                                               | (12.4) | [3.5]           |
| IM7/X2 <sup>2</sup> | 1426                                                                              | (8.1)  | [3.5] | 1726                                                                                       | (9.8)  | [3.5]           | 1920                                                                                               | (10.9) | [2.4]           |
| IM7/X2 <sup>3</sup> | 1268                                                                              | (7.2)  | [5.9] | 1691                                                                                       | (9.6)  | [8.4]           | 1761                                                                                               | (10.0) | [6.3]           |
| IM7/X3              | 1374                                                                              | (7.8)  | [7.8] | 1814                                                                                       | (10.3) | [19.7]          | 1462                                                                                               | (8.3)  | [27.1]          |
| IM7/X4              | 1532                                                                              | (8.7)  | [5.6] | 2078                                                                                       | (11.8) | [6.9]           | 2078                                                                                               | (11.8) | [4.3]           |
| AS4/3501-6          | 740                                                                               | (4.2)  | [7.1] | 916                                                                                        | (5.2)  | [5.3]           | 463                                                                                                | (2.6)  | [7.0]           |
| AS6/8552            | 617                                                                               | (3.5)  | [6.1] | 1532                                                                                       | (8.7)  | NA <sup>4</sup> | 1391                                                                                               | (7.9)  | NA <sup>4</sup> |

<sup>1</sup>CV (Coefficient of Variation, % based on five (5) replicates for each test

<sup>2</sup>IM7/X2 at 39% resin content (by weight)

<sup>3</sup>IM7/X2 at 32% resin content (by weight)

<sup>4</sup> average of two samples

### Compression After Impact

Compression After Impact (CAI) testing is often used for early screening assessment of performance of materials [2-4,7,18,19]. In this test, a quasi-isotropic panel (102 mm x 152 mm X 4.5 mm) thick is impacted with 667 N-m/(thickness) impact energy at low velocity (approximately 2.5 m/s) with a 12.7 mm diameter tup. This test is a combined damage resistance and damage tolerance test wherein damage is introduced during the impact event and subsequently tested in an end loaded compression fixture with semi-clamped

ends. A schematic of the CAI test setup is shown in Figure 8. A plot of CAI performance versus the Average Dynamic  $G_{IIC}$  presented in Table 2 is shown in Figure 9. The high and low data points are shown with error bars.

The CAI test is primarily a damage resistance test. The materials given in Table 1 have similar static damage tolerance performance based on open hole compression (OHC) tests and compression tests of laminates with similar impact damage [18]. That is, given an equivalent, pre-existing damage state, the materials deliver the same static strength. The average OHC performances for the X8553, 8551-7, X-Series, 3501-6, and 8552 based laminates are 300 MPa, 293 MPa, 287 MPa, 300 MPa, and 320 MPa, respectively. However, the materials in Table 2 exhibit dramatically different damage resistance performance under dynamic conditions such as low velocity impact. Specifically, the static  $G_{IIC}$  data presented in Table 2 for the 3501-6 based laminate is higher than the 8552 laminate, but this situation is reversed under dynamic conditions. Thus, the Average Dynamic  $G_{IIC}$  appears to be a leading indicator for a preliminary assessment of the advanced composite laminates subjected to lateral impact. It is a measurement of the average strain energy release rate for Mode II fracture during delamination formation under dynamic loading conditions.

### Conclusions and Recommendations

In this study, an assessment of Mode I and Mode II fracture under static and dynamic conditions has been developed. Rate had little effect for the rates and materials tested in Mode I in this study. The rates tested are typical of the impact duration of low velocity impact [1]. Higher testing rates may show a stronger effect. In contrast, the Mode II fracture performance of toughened systems is considerably better than homogeneous, brittle systems. While the apparent dynamic fracture properties of the systems are higher under dynamic conditions, the fracture performance of the toughened multi-phase systems appears to be less strain rate sensitive than the homogeneous systems. These results establish the need to examine dynamic fracture properties of materials for assessment of damage resistance during low velocity impact.

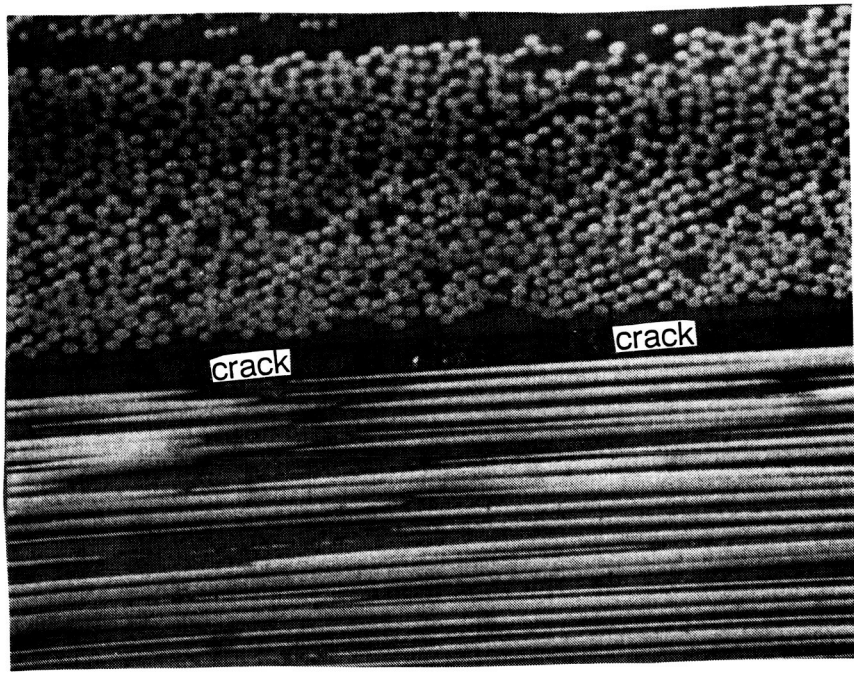
The simple, dynamic Mode II fracture test presented herein is somewhat easier to run, and considerably less expensive than CAI testing. Also, it has more meaning since it provides a measurement of a basic fracture mode, rather than a combined damage resistance, damage tolerance and structural test. The strong trend presented in Figure 9 should not be taken as a panacea, as the ultimate goal of increased longevity of composite materials is a combined damage resistance and damage tolerance problem. A combination of tests under static and dynamic conditions is necessary to determine the longevity of a material and structure.

Future tests should include a wider variety of materials, including thermoplastics, which are known to be quite strain rate sensitive [5]. In addition, a study varying the specimen geometry, loading conditions and impactor metrics should be conducted to isolate the influence of structural and material parameters of fracture performance.

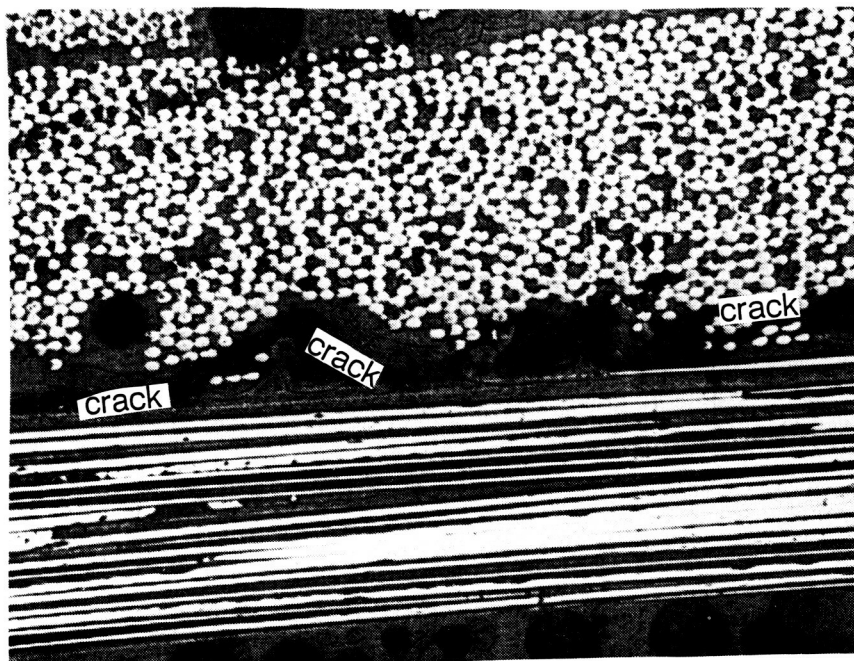
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a) AS4/3501-6 (self-similar crack at ply/matrix interface)



b) IM7/8551-7 (non self-similar, tortuous crack in interply process zone)

Figure 1. Crack Propagation in Hercules a) AS4/3501-6 and b) IM7/8551-7

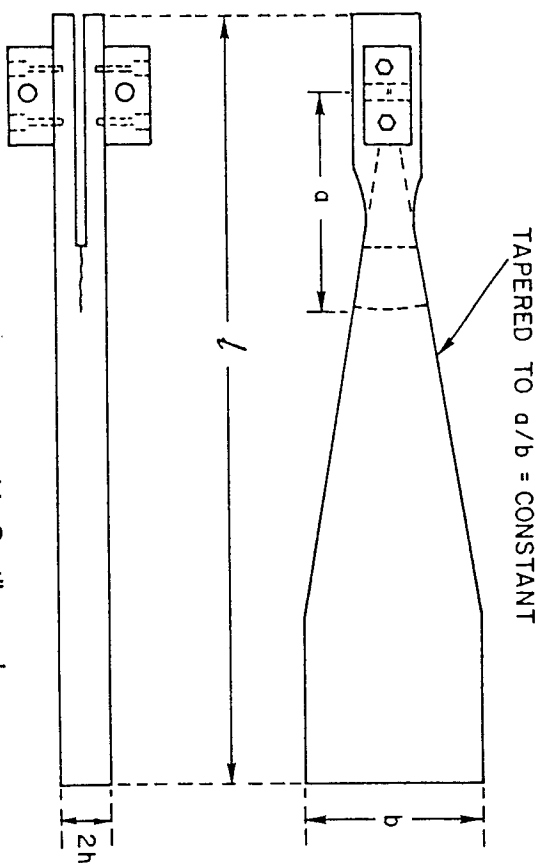


Figure 2. Width Tapered Double Cantilevered Beam (WTD CB) Geometry

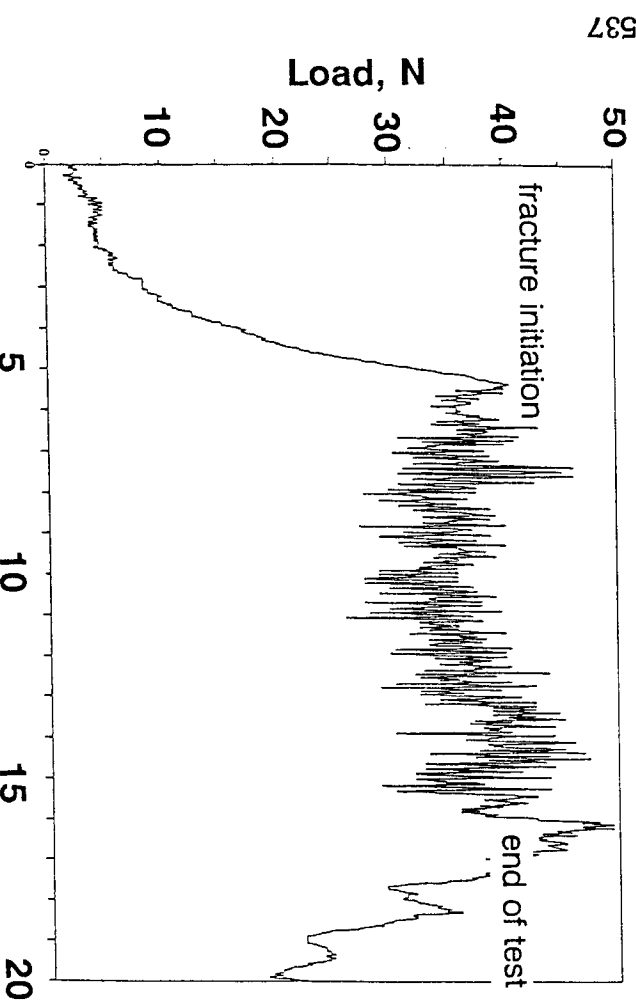


Figure 4. Load versus Time for Dynamic Mode I Fracture Testing

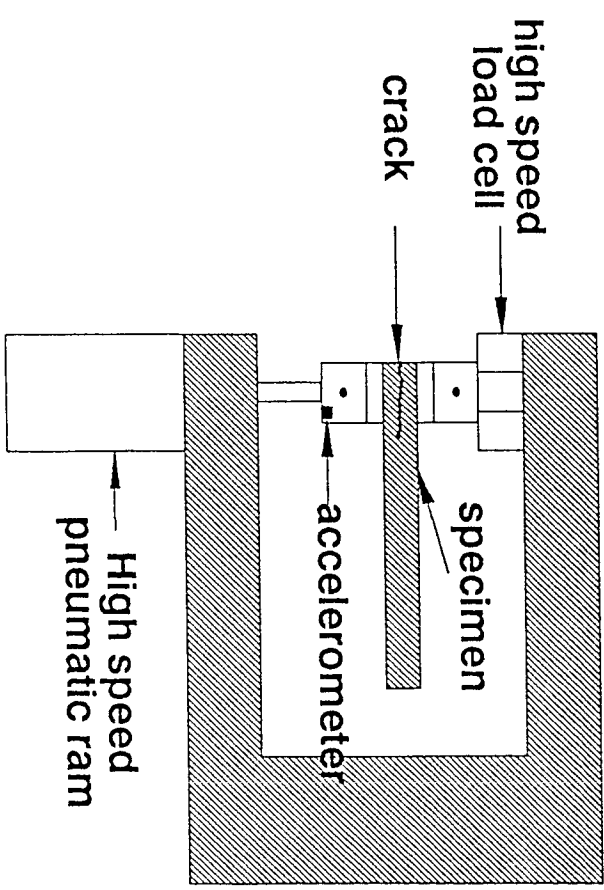


Figure 3. Test Setup for Dynamic  $G_{Ic}$  Testing

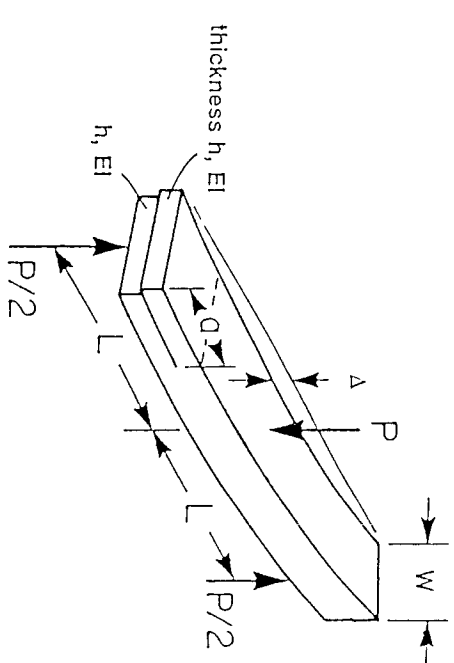


Figure 5. End Notched Flexure (ENF) Geometry

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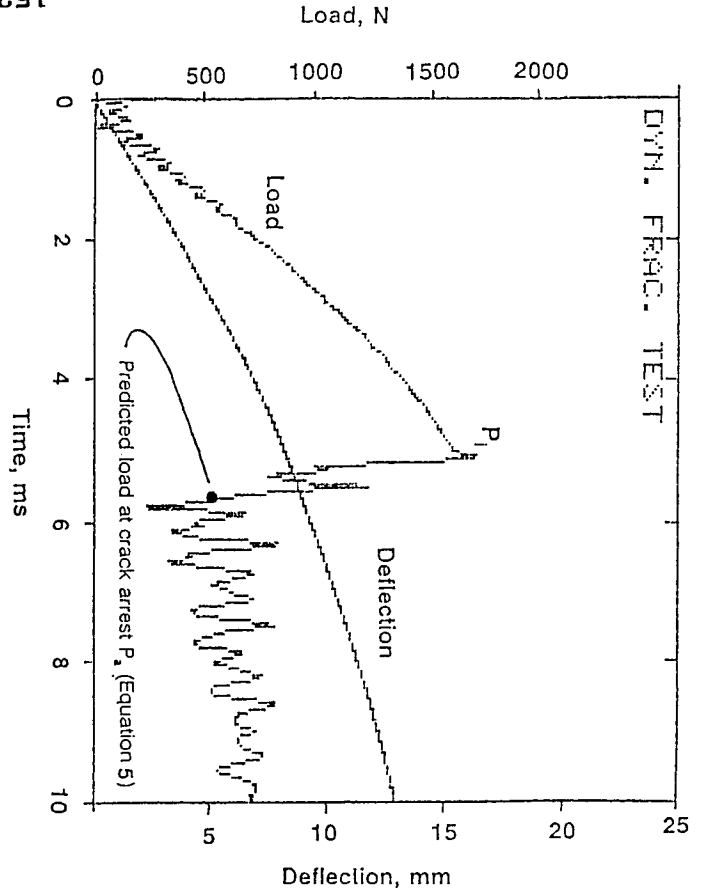


Figure 6. Load versus Time for Dynamic Mode II Fracture Testing

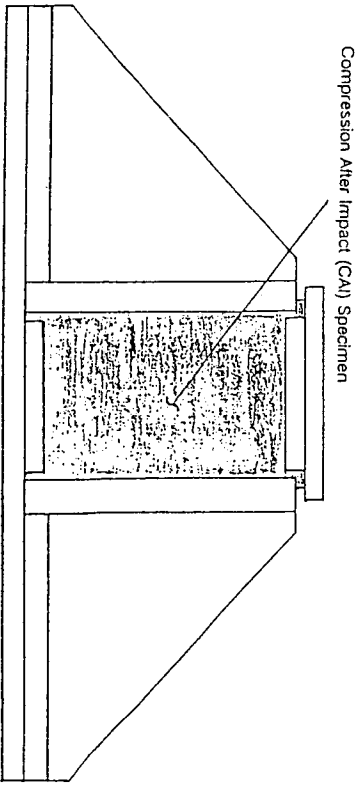


Figure 8. Schematic of Compression After Impact (CAI) Test Setup

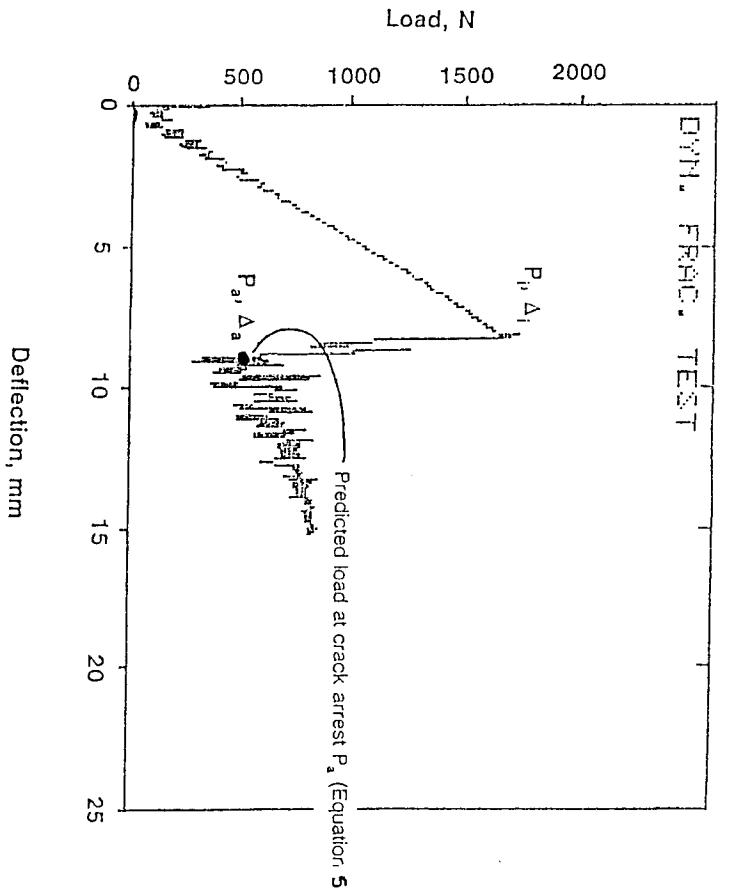


Figure 7. Load versus Deflection for Dynamic Mode II Fracture Testing

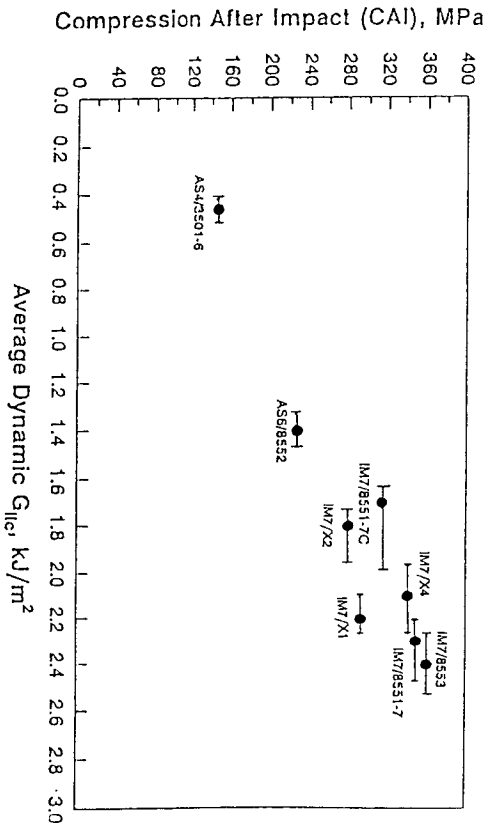


Figure 9. Compression After Impact Performance versus Average Dynamic  $G_{IIc}$

ANALYSIS TECHNIQUES FOR THE PREDICTION OF SPRINGBACK  
IN FORMED AND BONDED COMPOSITE COMPONENTS

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### SUMMARY

Two finite element analysis codes are used to model the effects of cooling on the dimensional stability of formed and bonded composite parts. The two analysis routines, one h-version and one p-version, are compared for modeling time, analysis execution time, and exactness of solution as compared to actual test results. A recommended procedure for predicting temperature effects on composite parts is presented, based on the results of this study.

### INTRODUCTION

McDonnell Aircraft Company is actively involved in the research and development of advanced composite structures. The goals of lighter weight, lower cost, and more survivable composite aircraft structures are being pursued by several research projects. Innovative designs are being studied in conjunction with emerging materials to realize these goals.

Several studies utilizing the thermoplastic composite material system are currently in progress. Design, manufacturing, and testing of sub-scale and full-scale components are part of the research activities. An interesting phenomenon which has arisen during fabrication and bonding is a dimensional warpage or "springback" of the sub and full scale thermoplastic components. This warpage is caused by the difference in the coefficients of thermal expansion (CTE) in the principal directions of the

thermoplastic material. As the part is cooled, the difference in the in-plane and through thickness CTE's causes a build-up of thermal stresses. The residual thermal stresses cause a warpage when the part is removed from the tool. In some instances, the part deflected off the tool at the edges as it was cooled. The resultant springback can occur in thermoplastic, thermoset, or any other composite material with different directional CTE's.

Related investigations have been completed to attempt to predict the amount of springback in formed or bonded composite parts. Material properties have been shown to be important in the prediction of springback magnitudes, references [1] and [2]. The through thickness coefficient of thermal expansion is an influential material property. Fahmy and Ragai-Ellozy derived a formula to estimate the through thickness CTE in reference [3]. The formula is based on known in-plane thermal properties.

Finite element solutions have been used to predict springback for simple geometries with some apparent success, references [4]-[6]. The majority of work has demonstrated that simple geometries can be modeled for springback with a reasonable degree of accuracy. However, more complex shapes are much more difficult to model.

Additional work is being completed in research activities at McDonnell Aircraft Company. The degree of springback has been shown to be linked to the build-up of residual thermal stresses. The effects of tooling temperature and ply orientation on the magnitude of these residual stresses are being investigated. The residual stresses have been shown to dramatically affect the interlaminar tensile strength of thermoplastic test specimens.

#### PROBLEM STATEMENT

The need exists to develop a reliable technique to predict springback in formed and bonded composite structures. Current procedures for springback compensation involve the reworking of

tools after the magnitude of springback is observed on a final part. This operation is costly and must often be repeated several times, depending on the complexity of the part. Tooling can be designed to compensate for springback if a reliable analysis method can be formulated. The compensated tool would then be capable of producing parts which are dimensionally correct the first time and eliminate the need for rework.

The use of finite element methods to predict springback has shown promise. H-version finite element codes have been used to model simple geometries with reasonable agreement with actual data. Lesser degrees of accuracy and reliability are realized for more complex geometries. The p-version finite element approach is an alternative to the current h-version methods. The p-version formulation increases the degree of polynomial shape function for each element as the mesh is refined. The number of elements remains constant and, therefore, requires less modeling time than an h-version analysis. Thus far, however, very little work has been done using p-version finite element codes to predict springback magnitudes.

This paper compares the h and p-version finite element codes for predicting springback magnitudes. Simple and complex geometries of isotropic and orthotropic material properties are modeled. The two analytical methods are then compared for modeling time, analysis execution time, and exactness of solution as compared to actual test results.

## DESCRIPTION OF ANALYSIS CODES

The two finite element codes chosen for comparison are COSTAR (h-version) and MSC/PROBE (p-version).

COSTAR, the COmposite STRuctural Analysis Routine, was developed by Jon Goering of the Structural Research Department of McDonnell Aircraft Company and is widely used in Research and Development projects. COSTAR elements are assembled and formulated

in a method similar to MSC/NASTRAN. Beam, shell, plate, and solid elements are available in linear, quadratic, and cubic forms. COSTAR is particularly well suited for the analysis of composite structures. The program has the ability to consider laminated and fully three-dimensional reinforced composites, as well as isotropic and orthotropic materials, reference [7]. The majority of figures in this paper were generated by the COSTAR post-processor.

MSC/PROBE uses the p-version approach to the finite element solution by increasing the degree of polynomial shape function of the element to the point of solution convergence. This feature often decreases modeling and solution time due to the requirement of fewer elements.

MSC/PROBE offers an error estimation routine in which the finite element solution is compared to the exact problem solution. The exact solution is based on the Principle of Virtual Work and is of the form;

(1)

in which  $\delta U$  is a bilinear expression for the virtual work of internal stresses and  $\delta W$  is a linear functional of the virtual work of external forces. Error estimation is based on solving the equation such that the difference between the finite element solution and the exact solution, based on the Principle of Virtual Work, is minimized. This can be expressed as;

(2)

in which  $U$  is the exact solution of Equation (1) and  $U_h$  is the user defined finite element solution. An excellent source of information on the formulations and methodology used in MSC/PROBE can be found in reference [8].

Other features such as the ability to calculate stress intensity factors around singularity points are available. The

author would refer readers to the MSC/PROBE User's Manual for detailed descriptions of other capabilities, reference [9].

## BASELINE COMPARISONS

### CASE I: SIMPLE GEOMETRY - ISOTROPIC MATERIAL

The initial test of the ability of COSTAR and MSC/PROBE to model springback was a simple isotropic hat section, Figure 1. The material properties used are as follows:

$$\begin{aligned} E &= 0.5 \times 10^6 \text{ psi} & \text{NU} &= 0.35 \\ \text{CTE} &= 0.31 \times 10^{-6} \text{ } 1/^{\circ}\text{F} \end{aligned}$$

The model discretization required for the two programs can be seen in Figures 2a and 2b. The COSTAR model required 388 elements and 980 degrees of freedom while the MSC/PROBE model required 32 elements and 1048 degrees of freedom at a p-level of 5. The applied load was a  $-500^{\circ}\text{F}$  temperature gradient to represent the cooldown of the part from consolidation temperature. The constrained model is shown in Figure 1. These load and constraint conditions were used on all models for the duration of the study.

The displacement results for the simple geometry, isotropic case can be seen in Table 1. As can be seen, COSTAR and MSC/PROBE give the same displacement results for this configuration. The error estimation for MSC/PROBE p-levels 1-8 is seen in Table 2. The execution time for the two analyses is very similar for the single COSTAR run and the MSC/PROBE run for  $p = 5$  only. The final deflected geometry can be seen in Figure 3.

### CASE II: SIMPLE GEOMETRY - ORTHOTROPIC MATERIAL

The effect of material property on the results of the two analysis programs was then studied. The isotropic material of Case

I was changed to the following orthotropic properties:

$$\begin{array}{ll} E_{11} = 2.24 \times 10^7 \text{ psi} & \nu_{12} = 0.35 \\ E_{22} = 1.30 \times 10^6 \text{ psi} & \nu_{23} = 0.48 \\ E_{33} = 1.30 \times 10^6 \text{ psi} & \nu_{31} = 0.02 \end{array}$$

$$\begin{array}{ll} CTE_{11} = 0.10 \times 10^{-6} \text{ } 1/^{\circ}\text{F} & G = 0.79 \times 10^6 \text{ psi} \\ CTE_{22} = 0.165 \times 10^{-4} \text{ } 1/^{\circ}\text{F} & \\ CTE_{33} = 0.175 \times 10^{-4} \text{ } 1/^{\circ}\text{F} & \end{array}$$

All other model parameters were identical to Case I. The results again show very close agreement between the two models, Table 3. Error estimations for the Case II model are similar to the Case I calculations seen in Table 2. The deflected geometry can be seen Figure 4.

#### CASE III: REFINED MESH - ORTHOTROPIC MATERIAL

The effect of mesh discretization on the results of the MSC/PROBE model was demonstrated by performing an h-p extension at the two radii. The number of elements was increased to 40 which can be seen in Figure 5. The results of this analysis again show close agreement between the two programs. Table 4 shows the displacements calculated from this analysis which are nearly identical to the results of Case II, Table 3. The effect of the h-p extension on the energy norm error can be seen in Table 5. The calculated error between the exact solution and finite element solution is decreased for the same p-levels of Table 2.

#### CASES I - III RESULTS SUMMARY

The results of Cases I - III show nearly exact agreement between COSTAR and MSC/PROBE in the calculation of temperature induced displacements. Analysis execution times between a single COSTAR run and an MSC/PROBE p-level 5 run are similar. MSC/PROBE

provides an error estimation function which can be used to evaluate the accuracy of the MSC/PROBE model. An h-p extension on the MSC/PROBE model was shown to decrease the percent error in the energy norm for the simple hat geometry.

### STIFFENED SKIN ANALYSIS

The results of the previous case studies indicate that COSTAR and MSC/PROBE will predict similar springback magnitudes for a simple geometry. The current research problem, however, is the springback encountered in the bonding of a thermoplastic hat-stiffened skin. This more complex geometry is modeled in an attempt to accurately predict the expected amount of springback.

Material properties for the corrugated panel and the flat skin panel to be bonded were determined using the program THICKLAM, reference 10. THICKLAM utilizes Classical Lamination Theory to derive three-dimensional elastic constants of composite laminates using the formulae of Jones and Sun, references 11 and 12. The three-dimensional properties are necessary due to the cross sectional geometry of the model. The model X axis follows a material  $90^\circ$  ply and the model Y axis is through the thickness of the skin. Therefore E1 of the model is E<sub>y</sub> of the material and E2 of the model is E<sub>z</sub> of the material, Figure 6. The material properties used for the corrugation and flat skin panels are shown in Tables 6a and 6b.

The COSTAR and MSC/PROBE finite element models are shown in Figures 7a and 7b. The COSTAR model required approximately 1500 elements and 2325 degrees of freedom while the MSC/PROBE model required 153 elements and 3400 degrees of freedom for  $p = 4$ . The material properties of Table 6 were used for the corresponding parts of the model and the proper local material properties were input into the MSC/PROBE model. The applied load was a  $-500^\circ\text{F}$  temperature gradient as in the previous case studies. The constraints are again similar to the previous studies and are shown in Figure 6.



The results of this study again show close agreement between the two finite element programs, even for the more complex geometry. Table 7 shows the deflections for the two models. Figure 8 shows the deflected geometry of the COSTAR model which is essentially the same as the MSC/PROBE model. The more sophisticated model produced a greater error as can be seen in the calculations of Table 8. The COSTAR model required approximately one minute of execution time for the solution. The MSC/PROBE model required approximately forty minutes of execution time for all p-levels 1-8.

### SUMMARY OF RESULTS AND CONCLUSIONS

The COSTAR and MSC/PROBE finite element programs predict almost exactly the same magnitudes of springback for a range of geometries and materials. Less time was required to construct the MSC/PROBE models due to the fewer number of elements. The addition of the orthotropic material angle in MSC/PROBE Version 4.1 further reduces the amount of modeling time required.

A single execution of the COSTAR model was used throughout this project as a comparison to the eight p-levels solutions available with MSC/PROBE. This is an important factor due to the error encountered in the final stiffened skin model. The COSTAR model must be re-meshed in order to improve the solution. The MSC/PROBE model would require less extensive alterations and, therefore, requires less time to improve the solution.

The two programs show good agreement with actual results. A stiffened panel of the same dimensions and material properties as those modeled was previously fabricated and bonded. The total springback measured for the panel was 0.29 inches. The magnitude of the springback predicted by both COSTAR and MSC/PROBE is 0.22 inches for a difference of 24%.

### FUTURE WORK

The finite element methods used for this study have been shown

to predict the correct deflected geometry of a thermoplastic stiffened skin. Current research has shown that such factors as material properties and geometry have a great impact on the magnitude of springback in a composite part. Other physical and model parameters should be investigated in order to improve the predicted magnitudes of springback.

The bondline of the stiffened skin was thought to have little effect on springback and was not modeled. This may be an important omission from the model. Material properties, particularly the through thickness CTE, must be accurately determined for use in springback prediction. Thermoset materials experience a chemical shrinkage during cure which effectively increases the through thickness CTE. This effect must be accurately quantified if thermoset springback prediction is to be accomplished.

Geometry plays an important role in springback prediction. Current models require two dimensional representation only. Eventually this must be extended to three dimensions. Plane strain plate elements are used for the two dimensional cases and seem to be the most reliable. Other element formulations may prove to be more accurate. The type of element to be used for the three dimensional case has not been investigated.

The solution used for this project was for an applied load of a single temperature step. Material properties were held constant over the entire interval. The actual case would change the material properties as the part is cooled from the stress free state. A piecewise linear solution may be more realistic than the single interval used for this study. Several temperature steps should be applied to the model with material properties varied accordingly for each interval. This load and material property condition would more realistically model the actual behavior of the part as it cools.

#### RECOMMENDED APPROACH FOR SPRINGBACK PREDICTIONS

1. Verify all geometry and use on the finite element model as

accurately as possible.

2. Components with three-dimensional curvature should be sectioned into appropriate two-dimensional sections, if at all possible.

Solid element modeling of complex three-dimensional parts is possible, but the advantages over 2-D modeling are limited and normally not worth the additional modeling time.

3. Two-dimensional generalized plane strain elements are recommended. These elements seem to most closely model the actual stress/strain behavior of the material, particularly if the material is a composite.

4. At least five elements should be used to define a radius. Four elements through the thickness are necessary for an h-version finite element code while only one element is required for a p-version code.

5. Temperature dependent orthotropic properties should be used to define the materials. The properties should be broken into corresponding temperature intervals, the size of which will depend on the material and overall change in temperature.

6. The finite element model is then run for each of the defined temperature intervals with corresponding material properties. The final solution is the summation of the resulting deflections from each analysis run.

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**TABLE 1. DISPLACEMENT RESULTS FOR CASE I, ISOTROPIC MATERIAL**

| MSC/PROBE |         |          | COSTAR |          |          |
|-----------|---------|----------|--------|----------|----------|
| Node #    | Delta X | Delta Y  | Node # | Delta X  | Delta Y  |
| 1         | 0.0     | -0.0200  | 1      | 0.0      | -0.01999 |
| 5         | 0.0     | -0.212   | 5      | 0.0      | -0.02118 |
| 931       | -0.0128 | -0.00166 | 437    | -0.01288 | -0.00156 |
| 936       | -0.137  | -0.00242 | 441    | -0.01378 | -0.00234 |
| 911       | -0.0502 | 0.0      | 911    | -0.05022 | 0.0      |
| 915       | -0.0502 | -0.00120 | 915    | -0.05022 | -0.00197 |

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**TABLE 2. ERROR ESTIMATION FOR CASE 1, ISOTROPIC MATERIAL**

| P | Global DOF | Delta Y      | Extrapolated Energy | Convergence Rate | Percent Rel Error Est'd |
|---|------------|--------------|---------------------|------------------|-------------------------|
| 1 | 120        | 0.102792D-01 | 0.000000D+00        | 0.00             | *****                   |
| 2 | 304        | 0.490144D-05 | 0.000000D+00        | 9.65             | 60.25                   |
| 3 | 488        | 0.431920D-05 | 0.431515D-05        | 1.25             | 44.84                   |
| 4 | 736        | 0.408346D-05 | 0.385223D-05        | 0.96             | 36.81                   |
| 5 | 1048       | 0.401905D-05 | 0.398416D-05        | 0.40             | 34.29                   |
| 6 | 1424       | 0.381051D-05 | 0.356409D-05        | 2.22             | 24.41                   |
| 7 | 1864       | 0.377486D-05 | 0.376470D-05        | 0.68             | 22.29                   |
| 8 | 2368       | 0.370690D-05 | 0.359619D-05        | 2.00             | 17.55                   |

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**TABLE 3. DISPLACEMENT RESULTS FOR CASE II, ORTHOTROPIC MATERIAL**

| MSC/PROBE |         |          | COSTAR |          |          |
|-----------|---------|----------|--------|----------|----------|
| Node #    | Delta X | Delta Y  | Node # | Delta X  | Delta Y  |
| 1         | 0.0     | 0.00371  | 1      | 0.0      | 0.00372  |
| 5         | 0.0     | 0.00300  | 5      | 0.0      | -0.00301 |
| 931       | -0.0143 | 0.00082  | 437    | -0.01443 | -0.00078 |
| 936       | -0.0144 | -0.00010 | 441    | -0.01457 | -0.00012 |
| 911       | -0.0152 | 0.0      | 911    | -0.01533 | 0.0      |
| 915       | -0.0152 | -0.00071 | 915    | -0.01533 | -0.00071 |

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**TABLE 4. DISPLACEMENT RESULTS FOR  
CASE III, ORTHOTROPIC MATERIAL,  
REVISED MSC/PROBE MESH**

| MSC/PROBE |         |          |
|-----------|---------|----------|
| Node #    | Delta X | Delta Y  |
| 1         | 0.0     | 0.00372  |
| 5         | 0.0     | 0.00301  |
| 931       | -0.0148 | 0.00034  |
| 936       | -0.0148 | -0.00045 |
| 911       | -0.0153 | 0.0      |
| 915       | -0.0153 | -0.00071 |

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**TABLE 5. ERROR ESTIMATION FOR CASE III, REVISED MSC/PROBE MODEL, ORTHOTROPIC MATERIAL**

| P | Global DOF | Total Energy | Extrapolated Energy | Convergence Rate | Percent Rel Error Est'd |
|---|------------|--------------|---------------------|------------------|-------------------------|
| 1 | 152        | 0.682723D-02 | 0.000000D+00        | 0.00             | *****                   |
| 2 | 348        | 0.407940D-05 | 0.000000D+00        | 9.82             | 47.89                   |
| 3 | 616        | 0.391884D-05 | 0.391814D-05        | 0.50             | 42.54                   |
| 4 | 928        | 0.370865D-05 | 0.354308D-05        | 1.05             | 34.29                   |
| 5 | 1320       | 0.355625D-05 | 0.266208D-05        | 1.41             | 26.77                   |
| 6 | 1792       | 0.344546D-05 | 0.282333D-05        | 2.05             | 19.57                   |
| 7 | 2344       | 0.338463D-05 | 0.327377D-05        | 2.43             | 14.12                   |
| 8 | 2976       | 0.335552D-05 | 0.331843D-05        | 2.43             | 10.57                   |

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**TABLE 6a. MATERIAL PROPERTY CALCULATIONS FOR THE CORRUGATION**

THICKLAM assumes that the composite is a balanced, symmetric laminate of one material system.

**Input**

| Material System    | IM7/PEEK    |
|--------------------|-------------|
| Lamina Orientation |             |
| Angle 1            | 0°          |
| Angle 2            | 45°         |
| Angle 3            | -45°        |
| Angle 4            | 90°         |
| Angle 5            |             |
| Angle 6            |             |
| Angle 7            |             |
| Angle 8            |             |
| Lamina Data        |             |
| E <sub>1</sub>     | 2.24E+07    |
| E <sub>2</sub>     | 1.30E+06    |
| E <sub>3</sub>     | 1.30E+06    |
| μ <sub>12</sub>    | 0.350       |
| μ <sub>13</sub>    | 0.020       |
| μ <sub>23</sub>    | 0.480       |
| G <sub>12</sub>    | 7.90E+05    |
| G <sub>13</sub>    | 7.90E+05    |
| G <sub>23</sub>    | 4.50E+05    |
| μ <sub>21</sub>    | 0.020       |
| μ <sub>31</sub>    | 0.001       |
| μ <sub>32</sub>    | 0.480       |
| Δ                  | 2.01308E-20 |

| Temperature       | RT    |
|-------------------|-------|
| Moisture Content  | Dry   |
| Percent Thickness |       |
| Angle 1           | 30.9% |
| Angle 2           | 30.8% |
| Angle 3           | 30.8% |
| Angle 4           | 7.7%  |
| Angle 5           |       |
| Angle 6           |       |
| Angle 7           |       |
| Angle 8           |       |
| Total Thickness   | 0.07  |

**Output**

|                                    |          |
|------------------------------------|----------|
| Lamina (C) Matrix                  |          |
| C11                                | 2.26E+07 |
| C12                                | 6.14E+05 |
| C13                                | 3.21E+05 |
| C22                                | 1.71E+06 |
| C23                                | 8.20E+05 |
| C33                                | 1.69E+06 |
| C44                                | 4.50E+05 |
| C55                                | 7.90E+05 |
| C66                                | 7.90E+05 |
| Laminate Data                      |          |
| E <sub>x</sub> (E <sub>3</sub> )   | 9.46E+06 |
| E <sub>y</sub> (E <sub>1</sub> )   | 5.38E+06 |
| E <sub>z</sub> (E <sub>2</sub> )   | 1.63E+06 |
| μ <sub>xy</sub> (μ <sub>31</sub> ) | 0.541    |
| μ <sub>xz</sub> (μ <sub>32</sub> ) | 0.102    |
| μ <sub>yz</sub> (μ <sub>12</sub> ) | 0.278    |
| G <sub>xy</sub> (G <sub>31</sub> ) | 3.86E+06 |
| G <sub>xz</sub> (G <sub>32</sub> ) | 6.12E+05 |
| G <sub>yz</sub> (G <sub>12</sub> ) | 5.39E+05 |
| μ <sub>yx</sub> (μ <sub>13</sub> ) | 0.307    |
| μ <sub>zx</sub> (μ <sub>23</sub> ) | 0.018    |
| μ <sub>zy</sub> (μ <sub>21</sub> ) | 0.084    |

GP14-0169-16-D/rr



**TABLE 6b. MATERIAL PROPERTY CALCULATIONS FOR THE FLAT SKIN PANEL**

THICKLAM assumes that the composite is a balanced, symmetric laminate of one material system.

**Input**

|                    |             |
|--------------------|-------------|
| Material System    | IM7/PEEK    |
| Lamina Orientation |             |
| Angle 1            | 0°          |
| Angle 2            | 45°         |
| Angle 3            | -45°        |
| Angle 4            | 90°         |
| Angle 5            |             |
| Angle 6            |             |
| Angle 7            |             |
| Angle 8            |             |
| Lamina Data        |             |
| E <sub>1</sub>     | 2.24E+07    |
| E <sub>2</sub>     | 1.30E+06    |
| E <sub>3</sub>     | 1.30E+06    |
| μ <sub>12</sub>    | 0.350       |
| μ <sub>13</sub>    | 0.020       |
| μ <sub>23</sub>    | 0.480       |
| G <sub>12</sub>    | 7.90E+05    |
| G <sub>13</sub>    | 7.90E+05    |
| G <sub>23</sub>    | 4.50E+05    |
| μ <sub>21</sub>    | 0.020       |
| μ <sub>31</sub>    | 0.001       |
| μ <sub>32</sub>    | 0.480       |
| Δ                  | 2.01308E-20 |

|                   |       |
|-------------------|-------|
| Temperature       | RT    |
| Moisture Content  | Dry   |
| Percent Thickness |       |
| Angle 1           | 30.9% |
| Angle 2           | 30.8% |
| Angle 3           | 30.8% |
| Angle 4           | 7.7%  |
| Angle 5           |       |
| Angle 6           |       |
| Angle 7           |       |
| Angle 8           |       |
| Total Thickness   | 0.07  |

**Output**

|                                    |          |
|------------------------------------|----------|
| Lamina (C) Matrix                  |          |
| C11                                | 2.26E+07 |
| C12                                | 6.14E+05 |
| C13                                | 3.21E+05 |
| C22                                | 1.71E+06 |
| C23                                | 8.20E+05 |
| C33                                | 1.69E+06 |
| C44                                | 4.50E+05 |
| C55                                | 7.90E+05 |
| C66                                | 7.90E+05 |
| Laminate Data                      |          |
| E <sub>x</sub> (E <sub>3</sub> )   | 1.14E+07 |
| E <sub>y</sub> (E <sub>1</sub> )   | 7.37E+06 |
| E <sub>z</sub> (E <sub>2</sub> )   | 1.64E+06 |
| μ <sub>xy</sub> (μ <sub>31</sub> ) | 0.308    |
| μ <sub>xz</sub> (μ <sub>32</sub> ) | 0.194    |
| μ <sub>yz</sub> (μ <sub>12</sub> ) | 0.305    |
| G <sub>xy</sub> (G <sub>31</sub> ) | 2.78E+06 |
| G <sub>xz</sub> (G <sub>32</sub> ) | 6.07E+05 |
| G <sub>yz</sub> (G <sub>12</sub> ) | 5.44E+05 |
| μ <sub>yx</sub> (μ <sub>13</sub> ) | 0.200    |
| μ <sub>zx</sub> (μ <sub>23</sub> ) | 0.028    |
| μ <sub>zy</sub> (μ <sub>21</sub> ) | 0.068    |

GP14-0169-17-D/rr

**TABLE 7. DISPLACEMENT RESULTS FOR THE FULL STIFFENED SKIN**

| MSC/PROBE |          |         | COSTAR |          |         |
|-----------|----------|---------|--------|----------|---------|
| Node #    | Delta X  | Delta Y | Node # | Delta X  | Delta Y |
| 236       | -0.00952 | 0.0949  | 1      | -0.00728 | 0.10900 |
| 229       | -0.01100 | 0.0940  | 11     | -0.01196 | 0.10660 |
| 363       | -0.00580 | 0.0616  | 103    | -0.00564 | 0.06158 |
| 237       | -0.00793 | 0.0603  | 116    | -0.00637 | 0.05993 |

GP14-0169-18-D/cm

**TABLE 8. ERROR ESTIMATION FOR THE HAT-STIFFENED SKIN**

| P | Global DOF | Total Energy | Extrapolated Energy | Convergence Rate | Percent Rel Error Est'd |
|---|------------|--------------|---------------------|------------------|-------------------------|
| 4 | 3467       | 0.171003D+00 | 0.000000D+00        | 0.00             | 14.28                   |
| 5 | 4947       | 0.169596D+00 | 0.000000D+00        | 1.49             | 10.95                   |
| 6 | 6733       | 0.168854D+00 | 0.167585D+00        | 1.49             | 8.70                    |

GP14-0169-19-D/rr

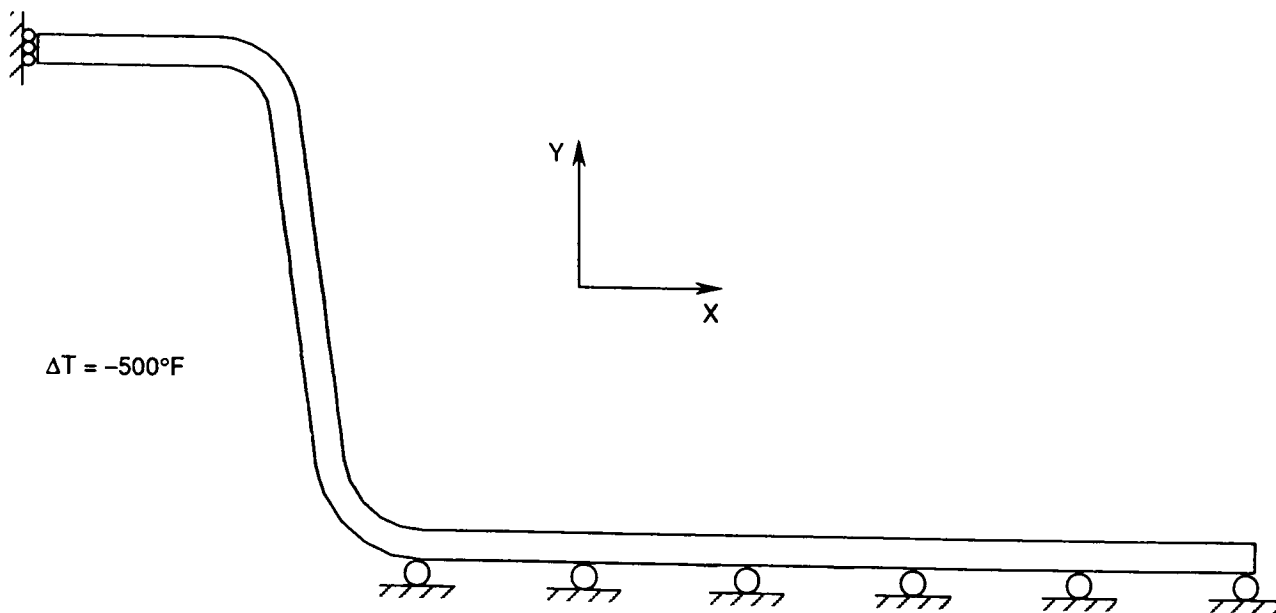


Figure 1. Simple Geometry Model for Baseline Cases I, II, and III

GP14-0169-1-D/kmg

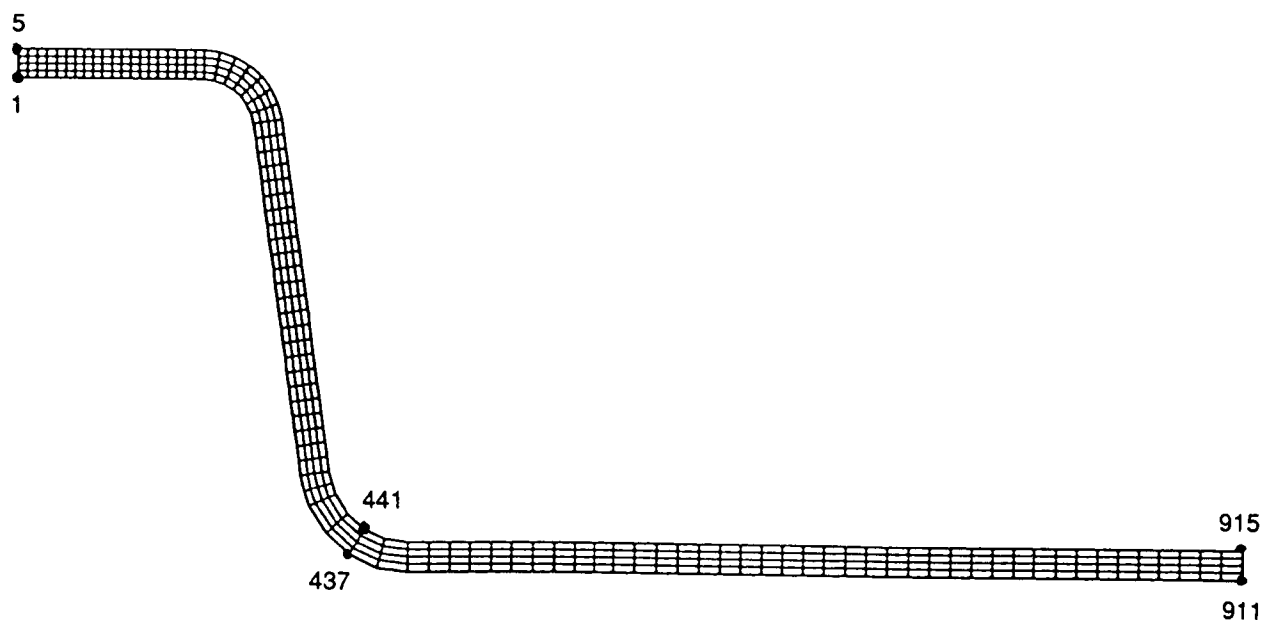
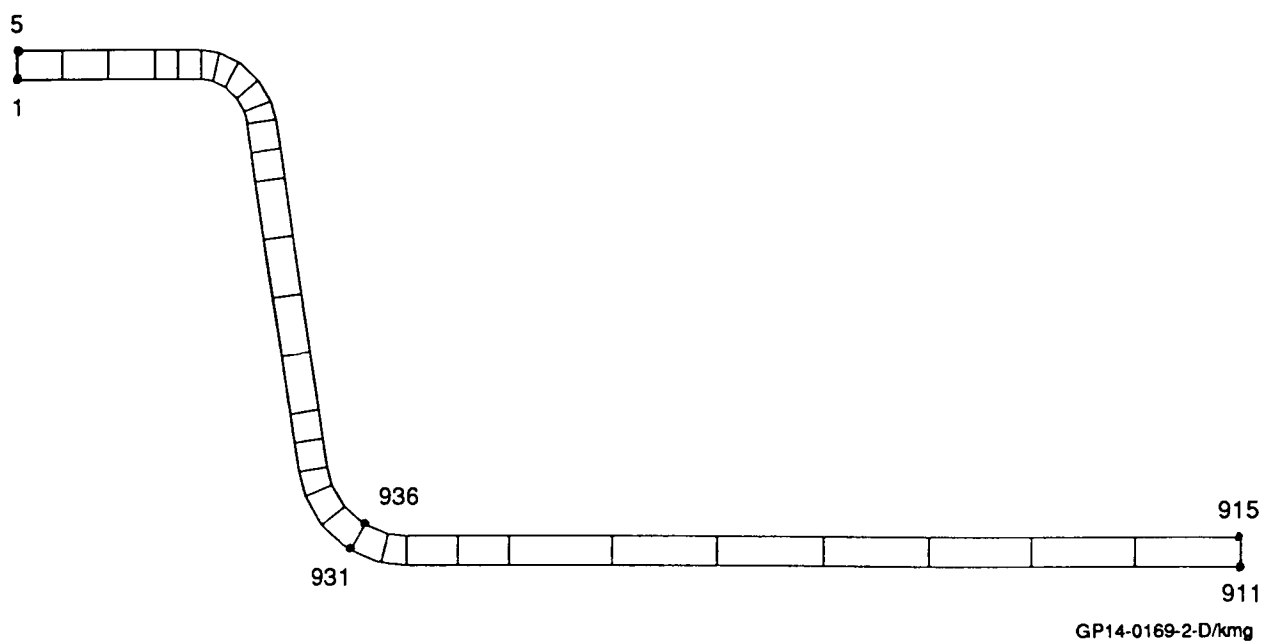
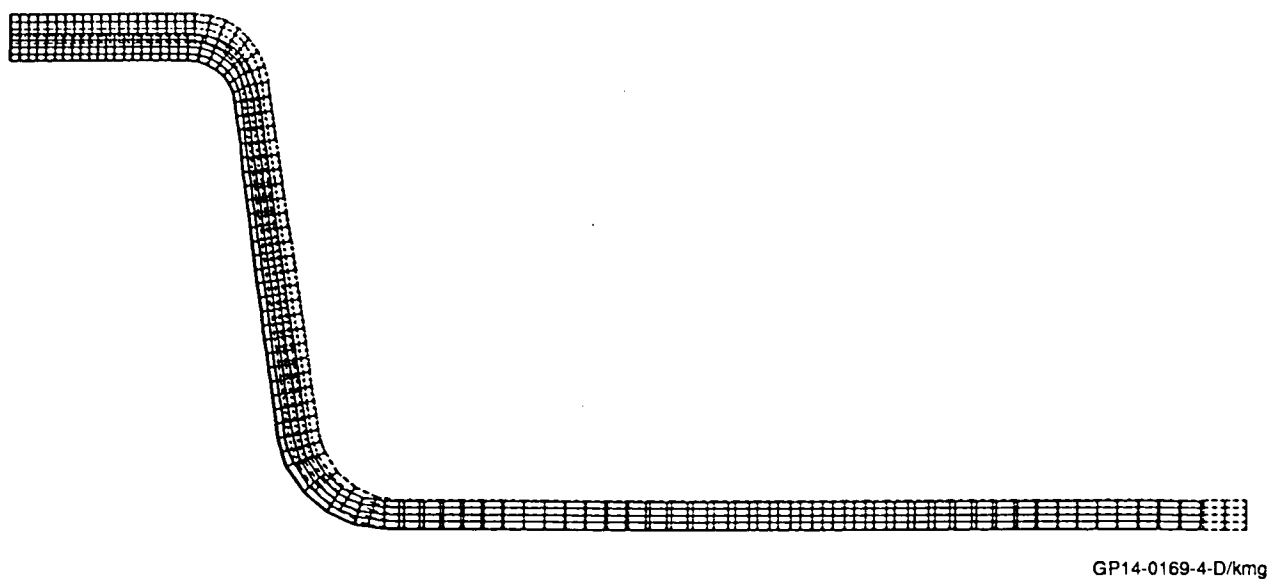


Figure 2a. Simple Geometry, COSTAR Model Discretization

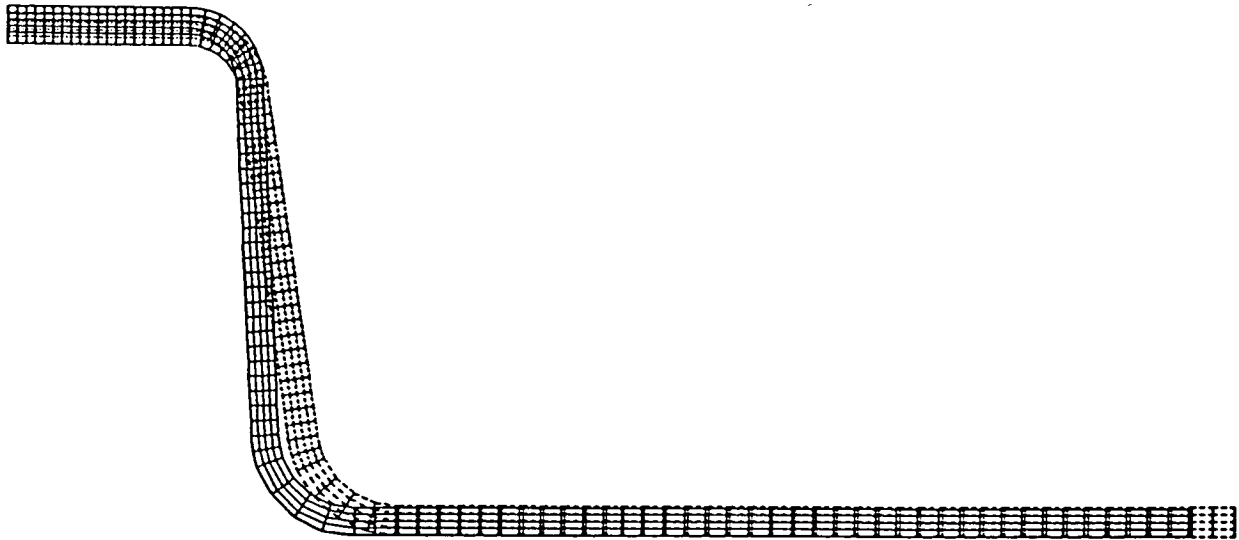
GP14-0169-3-D/kmg



**Figure 2b. Simple Geometry, MSC/PROBE Model Discretization**

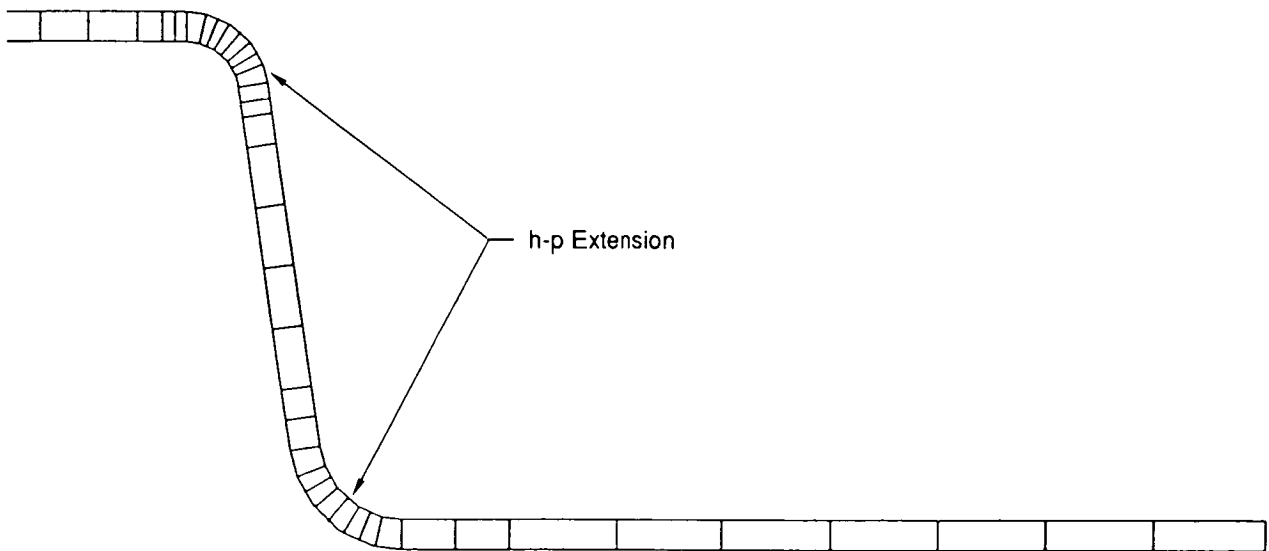


**Figure 3. Deflected Geometry for Case 1 Isotropic Material**



GP14-0169-5-D/kmg

**Figure 4. Deflected Geometry for Case II, Orthotropic Material**



GP14-0169-6-D/kmg

**Figure 5. MSC/PROBE Model for Case III, Refined Mesh**

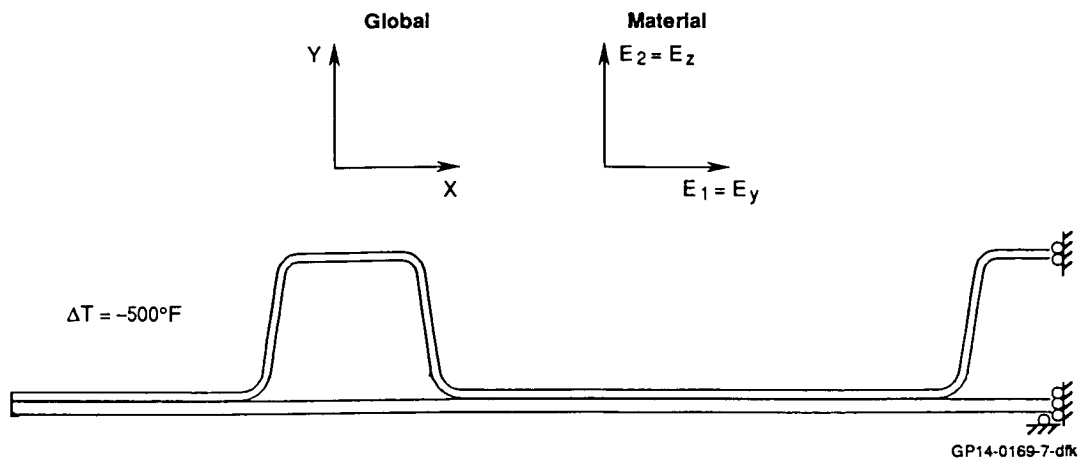


Figure 6. Geometry for the Full Hat-Stiffened Skin

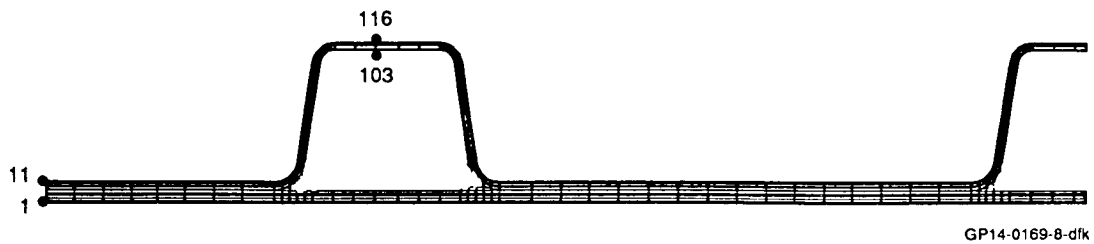


Figure 7a. COSTAR Model for the Hat-Stiffened Skin

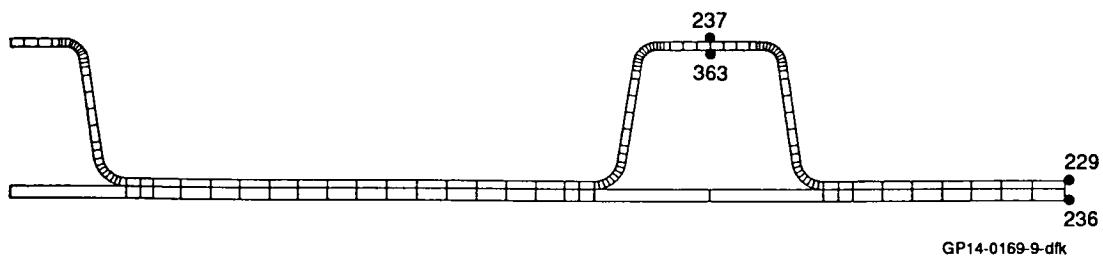


Figure 7b. MSC/PROBE Model for the Hat-Stiffened Skin

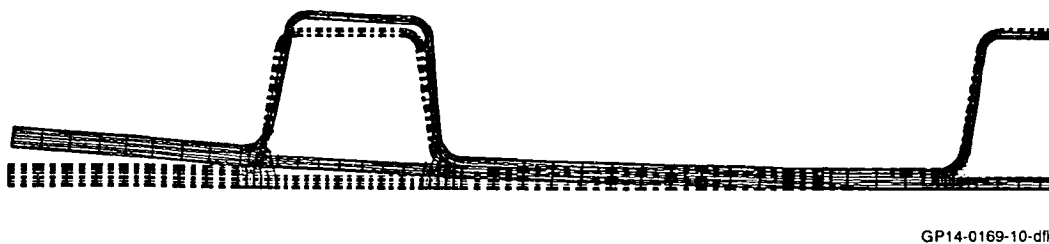


Figure 8. Deflected Geometry for the Hat-Stiffened Skin

**SESSION XI**  
**SPACE STRUCTURES**

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# STIFFNESS AND STRENGTH TAILORING IN UNIFORM SPACE-FILLING TRUSS STRUCTURES

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## SUMMARY

This paper presents a deterministic procedure for tailoring the continuum stiffness and strength of uniform space-filling truss structures through the appropriate selection of truss geometry and member sizes (i.e., flexural and axial stiffnesses and length). The trusses considered herein are generated by replication of a characteristic truss cell uniformly through space. The repeating cells are categorized by one of a set of possible geometric symmetry groups derived using the techniques of crystallography. The elastic symmetry associated with each geometric symmetry group is identified to aid in the selection of an appropriate truss geometry for a given application. Stiffness and strength tailoring of a given truss geometry is enabled through explicit expressions relating the continuum stiffnesses and failure stresses of the truss to the stiffnesses and failure loads of its members. These expressions are derived using an existing equivalent continuum analysis technique and a newly developed analytical failure theory for trusses. Several examples are presented to illustrate the application of these techniques, and to demonstrate the usefulness of the information gained from this analysis.

## INTRODUCTION

In the future, the primary structures of many large orbiting spacecraft will be lightweight trusses. Although numerous studies have been performed to determine the feasibility and structural characteristics of these trusses (e.g. refs. 1-3), little work has been done to establish deterministic procedures for their design. The selection of appropriate truss designs is influenced by both structural optimization and spacecraft operational considerations. Currently, structural optimization of these trusses is a predominantly heuristic process involving trial and error procedures. The purpose of this paper is to present a deterministic procedure for truss geometry selection and member design based on tailoring the continuum stiffness and strength characteristics of the truss. Analysis of the truss stiffness and strength characteristics is performed using an equivalent continuum analogy (ref. 4). This approach is preferred because it offers better insight into structural behavior than conventional numerical analysis techniques.

The trusses considered herein are generated by replication (rotational and/or translational) of a characteristic cell uniformly through space, as shown in figure 1, and are thus called uniform space-filling trusses. In most cases, the repeating truss cell and the resulting truss structure inherently possess some geometric symmetry. The presence of geometric symmetry implies elastic symmetry which reduces the number of independent equivalent elastic constants characterizing the truss. In this study, the techniques of crystallography are used to define the possible geometric symmetry groups associated with repeating cells which generate uniform trusses. In addition, the number of independent elastic constants associated with each geometric symmetry group is identified to aid in the selection of an appropriate truss geometry for a given application.

The independent elastic constants characterizing a truss can be tailored to specific values by selecting appropriate member stiffnesses. In the present study, this stiffness tailoring is accomplished using explicit relationships between the equivalent continuum stiffnesses of a truss and the axial stiffnesses of its members. Also, the continuum strength characteristics of a truss are tailored using a strength tensor which is written explicitly in terms of the local elastic buckling loads of the truss members. To illustrate the application of these techniques, a commonly used truss geometry is analyzed to determine member



sizes which produce optimum isotropic and orthotropic (i.e., one direction of high stiffness and strength) designs.

All derivations presented herein were performed symbolically using a computerized mathematics routine (ref. 5), and results were converted into a numerical form when necessary. The advantage in using symbolic algebra is that explicit relationships can be determined between the design parameters and the continuum elastic behavior of the truss. This significantly enhances the utility of the stiffness and strength tailoring procedures presented herein.

## SYMBOLS

|                        |                                                                                 |
|------------------------|---------------------------------------------------------------------------------|
| $A$                    | cross-sectional area of members in the regular Octahedral truss                 |
| $A_o$                  | cross-sectional area of members in the octahedral lattice of the Warren truss   |
| $A_c$                  | cross-sectional area of members in cubic lattice of the Warren truss            |
| $A_n$                  | cross-sectional area of members in $n$ th group                                 |
| $c_{mn}$               | continuum elastic constants (matrix form)                                       |
| $C_{ijkl}$             | continuum elastic constants (tensor form)                                       |
| $C'_{mnop}$            | transformed continuum elastic constants                                         |
| $(C'_{1111})_n$        | continuum unidirectional stiffness for $n$ th group of parallel members         |
| $E$                    | Young's modulus of truss material                                               |
| $E_{eq}$               | equivalent continuum Young's modulus                                            |
| $(E_{eq})_z$           | equivalent z-direction Young's modulus                                          |
| $(E_{eq})_{iso.}$      | equivalent Young's modulus of isotropic Warren truss                            |
| $G_{eq}$               | equivalent continuum shear modulus                                              |
| $L$                    | characteristic dimension of truss repeating cell                                |
| $l_n$                  | length of members in $n$ th group                                               |
| $r_n$                  | radius of gyration of members in $n$ th group                                   |
| $s_{mn}$               | continuum compliance constants (matrix form)                                    |
| $S_{ijkl}$             | continuum compliance constants (tensor form)                                    |
| $T_{ij}$               | coordinate transformation tensor                                                |
| $v_n$                  | volume fraction of $n$ th group of parallel members                             |
| $x, y, z$              | Cartesian coordinates                                                           |
| $\beta$                | length ratio of repeating truss cell in z direction                             |
| $\delta_n$             | ratio of cross-sectional area of members in $n$ th group to that of first group |
| $\epsilon_{ij}$        | strain tensor                                                                   |
| $(\epsilon_{crit.})_n$ | critical axial strain for $n$ th group of members                               |
| $\nu_{eq}$             | equivalent Poisson's ratio                                                      |
| $\phi_i$               | direction cosine with the $i$ th coordinate axis                                |
| $\rho$                 | density of truss material                                                       |

|                         |                                                |
|-------------------------|------------------------------------------------|
| $\rho_{eq}$             | equivalent continuum density                   |
| $\sigma_{ij}$           | stress tensor                                  |
| $\sigma_{ult}$          | continuum compression strength                 |
| $(\sigma_{ult})_z$      | z-direction compression strength               |
| $(\sigma_{ult})_{iso.}$ | compression strength of isotropic Warren truss |
| $[\Omega_{kl}]_n$       | strength tensor                                |
| $\theta, \varphi$       | spherical coordinates                          |

## TRUSS GEOMETRY SELECTION

The design of a truss is often governed by considerations other than the structural performance (e.g. ref. 6). For example, operational concerns such as the arrangement and integration of spacecraft subsystems onto a truss might dictate a particular geometry for the truss repeating cell. For applications in which operational concerns do not dominate, it is prudent to select a truss geometry by matching its inherent elastic behavior with the structural requirements of the spacecraft. Even in situations where operational concerns prevail, it is probable that enough latitude exists in the selection of a truss geometry that structural considerations can be incorporated. The purpose of this section is to categorize the elastic characteristics of most uniform space-filling truss structures by examining their geometric symmetry.

The uniform truss structures considered herein are similar to crystalline lattices since they both can be generated by replication of a characteristic repeating cell which typically possesses geometric symmetry. Of interest are symmetry with respect to specific rotations about one or more axes, and/or symmetry with respect to reflection about one or more planes. Symmetry in the truss geometry (i.e., lattice arrangement and member designs) implies symmetry in the elastic characteristics of the truss. This implied elastic symmetry reduces the number of independent equivalent elastic constants characterizing the continuum behavior of the truss, and thus simplifies the task of stiffness and strength tailoring.

### Rotational Symmetry Groups

Studies in crystallography (refs. 7,8) have shown that the rotational and reflectional symmetries in reticulated, or discrete, structures are limited to a set of 32 possible combinations which are commonly called crystallographic symmetry groups. Love (ref. 9) determined that the elastic behavior of most crystallographic symmetry groups can be derived by considering only rotational symmetry. For brevity, the few cases in which reflectional symmetry is important are not considered herein. By neglecting reflectional symmetry, the 32 crystallographic symmetry groups reduce to the ten rotational symmetry groups shown in figure 2.

Each symmetry group in figure 2 is identified by a specific combination of axes about which there is rotational symmetry. The orientations of these axes are shown relative to a Cartesian coordinate system, and the order of rotational symmetry is given by one of four graphical symbols: a cusped oval, a triangle, a square, or a hexagon. These symmetry symbols are related to the order of symmetry in the accompanying key. The order of symmetry is defined as  $n$ -gonal where the rotation angle is  $2\pi/n$  and  $n$  is either 2, 3, 4, or 6. Notice that in symmetry groups  $i$  and  $j$ , the trigonal symmetry axes lie along lines connecting the center of a cube with its corners, thus structures of these symmetry groups are often referred to as "cubic" structures.

Symmetry groups which possess more than one axis of rotational symmetry are called multiaxial. The three rotational symmetry axes presented for each of the multiaxial groups are not the only symmetry

axes possessed by those groups. A complete set can be generated by applying the symmetry operation of each axis to the others. For example in symmetry group  $d$ , applying trigonal symmetry about the  $z$  axis identifies four additional digonal symmetry axes in the  $x$ - $y$  plane separated by  $60^\circ$ .

Any truss structure which possesses axes of rotational symmetry can be categorized by one of the ten rotational symmetry groups in figure 2. This classification is accomplished by identifying all rotational symmetry axes within the structure, and then selecting a Cartesian coordinate system relative to these axes which matches one of the given symmetry groups. Once the symmetry group of the truss is identified, its inherent elastic behavior is determined using the methods which follow.

### Elastic Characteristics of Rotational Symmetry Groups

A uniform truss structure can be represented by an equivalent homogeneous anisotropic continuum characterized by 21 empirical elastic constants. These elastic constants appear as stiffnesses,  $c_{mn}$  or  $C_{ijkl}$ , in the constitutive equations given in equation (1a) in matrix form and equation (1b) in tensor form.

$$\begin{pmatrix} \sigma_{11} \\ \sigma_{22} \\ \sigma_{33} \\ \sigma_{23} \\ \sigma_{13} \\ \sigma_{12} \end{pmatrix} = \begin{bmatrix} c_{11} & c_{12} & c_{13} & c_{14} & c_{15} & c_{16} \\ c_{12} & c_{22} & c_{23} & c_{24} & c_{25} & c_{26} \\ c_{13} & c_{23} & c_{33} & c_{34} & c_{35} & c_{36} \\ c_{14} & c_{24} & c_{34} & c_{44} & c_{45} & c_{46} \\ c_{15} & c_{25} & c_{35} & c_{45} & c_{55} & c_{56} \\ c_{16} & c_{26} & c_{36} & c_{46} & c_{56} & c_{66} \end{bmatrix} \begin{pmatrix} \epsilon_{11} \\ \epsilon_{22} \\ \epsilon_{33} \\ 2\epsilon_{23} \\ 2\epsilon_{13} \\ 2\epsilon_{12} \end{pmatrix} \quad (1a)$$

$$\sigma_{ij} = C_{ijkl} \epsilon_{kl} \quad (1b)$$

When the truss possesses geometric symmetry, elastic symmetry is implied which reduces the number of independent continuum elastic constants.

A continuum which possesses geometric symmetry with respect to a linear orthogonal transformation  $T_{ij}$ , such as a rotation or a reflection, also possesses symmetry in its elastic constants (see for example ref. 10). Therefore, the transformed stiffness tensor  $C'_{ijkl}$  must be identical to the original tensor  $C_{ijkl}$ . Hence,

$$C'_{ijkl} = C_{mnop} T_{im} T_{jn} T_{ko} T_{lp} = C_{ijkl} \quad (2)$$

The number of independent elastic constants associated with each symmetry group, presented in figure 2, is determined using equation (2). A transformation tensor  $T_{ij}$  is determined for the specified rotation about each symmetry axis, and substituted into equation (2) to give 21 conditions on the stiffnesses  $C_{ijkl}$ . Some of these conditions are identically satisfied, whereas others can be satisfied only by the elimination or restriction of certain elastic constants. This process is repeated for all rotational symmetry axes in the given symmetry group, and the resulting reduced set of elastic constants defines the continuum elastic characteristics of any truss structure which is a member of that symmetry group.

For example, the independent elastic constants characterizing trusses of symmetry group  $a$  are determined by enforcing elastic symmetry with respect to a  $180^\circ$  rotation about the  $z$  axis. The transformation matrix for this rotation is

$$T_{ij} = \begin{bmatrix} -1 & 0 & 0 \\ 0 & -1 & 0 \\ 0 & 0 & 1 \end{bmatrix} \quad (3)$$

Substitution of equation (3) into equation (2) gives the following result

$$C_{ijkl} = \begin{cases} C_{ijkl} & , \text{ if an even number (or none) of the indices are 3} \\ -C_{ijkl} & , \text{ if an odd number of the indices are 3} \end{cases} \quad (4)$$

Satisfying the second condition in equation (4) requires the following to be true (note that, due to symmetry in  $C_{ijkl}$ , many possible permutations of the subscripts have been omitted).

$$\begin{aligned} C_{1123} = C_{1113} = C_{2223} = C_{2213} = C_{3323} = \\ C_{3313} = C_{2312} = C_{1312} = 0 \end{aligned} \quad (5)$$

Employing the usual conversion from tensor to matrix form (ref. 10), the following equivalent conditions exist for the components of the stiffness matrix.

$$c_{14} = c_{15} = c_{24} = c_{25} = c_{34} = c_{35} = c_{46} = c_{56} = 0 \quad (6)$$

Similar calculations can be made for the remaining symmetry groups in figure 2. Without presentation of the details, the conditions on continuum stiffnesses as well as the number of independent elastic constants for each symmetry group are presented in table I. A similar derivation shows that the conditions presented in table I must also be obeyed by the components of the continuum compliance tensor.

An obvious conclusion to be drawn from table I is that the presence of any symmetry in a truss lattice significantly reduces the number of independent elastic constants characterizing its continuum behavior. This result greatly simplifies the task of tailoring the stiffness and strength of most trusses. It should also be cautioned that the conditions on the elastic constants presented in table I are valid only for the coordinate axes presented in figure 2. For example, symmetry groups *b, f, g, h, i, and j* are indicated to have zero shear coupling stiffnesses (e.g.,  $c_{14}$ ,  $c_{15}$ , and  $c_{16}$ ) in the given coordinate system, but they might have non-zero coupling stiffnesses in an alternate coordinate system. As explained by Rosen (ref. 11), and seen in table I, none of the permissible geometric symmetry groups possess sufficient symmetry to insure isotropic elastic behavior. However, it will be shown that isotropy can be obtained by tailoring the relative stiffnesses of different truss members.

The information in table I should help in the selection of appropriate truss geometries for particular truss applications, and in the determination of additional stiffness tailoring requirements for the selected truss geometry. For example, if the primary loads in a truss are expected to occur in only one direction, it is most efficient to consider geometries which have less symmetry and can easily be tailored to have significantly higher stiffnesses and strengths in that direction (i.e., an orthotropic design). However, for a structure which may have to sustain loads in multiple directions, or one for which the loading conditions are not well defined, it may be best to consider truss geometries which possess more symmetry and can be tailored to behave isotropically.

## STIFFNESS AND STRENGTH TAILORING

Once a truss geometry has been selected, its independent elastic constants are identified using table I. The values of these constants can be adjusted for a particular application by tailoring the relative axial stiffnesses of the members comprising the truss. Likewise, changing the relative elastic buckling loads of different members alters the equivalent continuum strengths of the truss. Changing truss dimensions and member stiffnesses which do not violate the geometric symmetry of the truss allows the truss to remain in the same rotational symmetry group, thus the conditions on its continuum stiffnesses given in

table I remain valid. Alternatively, changing dimensions and member stiffnesses of a truss which violate its geometric symmetry changes its rotational symmetry group, thus altering the number of independent elastic constants characterizing its behavior. Stiffness and strength tailoring will be demonstrated later for a truss in which geometric symmetry is maintained and one in which geometric symmetry is altered.

### Equivalent Continuum Elastic Constants

Once a candidate truss for stiffness tailoring is selected, its continuum stiffnesses are calculated in terms of the axial stiffnesses of its members. The approach used in the recent study for calculating these stiffnesses was developed by Neyfeh and Hefzy (ref. 12) and can be thought of as a three-dimensional generalization of classical laminated plate theory (ref. 13) in which groups of parallel members within the truss are analogous to individual lamina. Since truss members carry only an axial load, each group of parallel members can be considered to form a unidirectional elastic continuum which has no transverse or shearing stiffnesses. The stiffnesses for the truss assemblage are obtained by summing the stiffnesses of each of the groups of parallel members. This superposition of stiffnesses implies that the continuum displacement field within a truss is single-valued which is consistent with the fact that truss members connected at a common point must have the same displacement at that point. Note that this is not the case for trusses with cross-laced members which can slide relative to one another, therefore it is cautioned that such designs should not be analyzed using the techniques of the present study.

Each group of parallel members is characterized by one non-zero equivalent stiffness which is in the local  $x'$  direction (the member longitudinal direction). This equivalent unidirectional stiffness is determined in equation (7) for the  $n$ th group of members.

$$(C'_{1111})_n = E v_n \quad (7)$$

where  $E$  is the Young's modulus of the material in the members and  $v_n$  is the volume fraction of the group of members (i.e., the ratio of the total volume of material in the members to the total volume of the truss).

The continuum stiffnesses for a truss are calculated by transforming the unidirectional stiffnesses for each of its groups of parallel members into a global coordinate system using equation (2), and summing the results as indicated by

$$C_{ijkl} = \sum_n (C'_{1111})_n (T_{1i} T_{1j} T_{1k} T_{1l})_n \quad (8)$$

Elements of the first row of the transformation tensor,  $T_{1i}$ , are simply the direction cosines between the longitudinal axis of the members and the  $i$ th coordinate axis. Therefore, equation (8) can be rewritten as

$$C_{ijkl} = \sum_n (C'_{1111})_n (\phi_i \phi_j \phi_k \phi_l)_n \quad (9)$$

where  $\phi_i$  are the direction cosines of the members. The continuum stiffnesses defined by equation (9) are explicit functions of the member extensional stiffnesses. This enables the translation of desired continuum stiffness characteristics into member axial stiffness tailoring rules.

Equation (9) produces additional restrictions on the continuum stiffnesses of uniform trusses which should be noted. Employing the usual conversion from the matrix form of the elastic constants to the tensor form (ref. 10), the values for the transverse and shear stiffnesses,  $c_{12}$  and  $c_{66}$  are found to be:

$$c_{12} = C_{1122} = \sum_n (C'_{1111})_n (\phi_1^2 \phi_2^2)_n \quad (10)$$

$$c_{66} = C_{1212} = \sum_n (C'_{1111})_n (\phi_1^2 \phi_2^2)_n \quad (11)$$

thus:

$$c_{12} = c_{66} \quad (12)$$

Similarly, it is found that

$$c_{13} = c_{55} \quad , \quad c_{23} = c_{44} \quad (13)$$

It is reiterated that these identities must hold for any uniform space-filling truss, regardless of its geometry, and therefore these identities should be added to those already presented in table I for all symmetry groups. Thus, under the assumptions above, a generally anisotropic space-filling truss structure has only 18 independent elastic constants rather than 21 as is normal for a generally anisotropic solid.

Trusses that are tailored to behave as isotropic continua can be characterized by two elastic constants, an equivalent Young's modulus,  $E_{eq}$ , and an equivalent Poisson's ratio,  $\nu_{eq}$ . Writing the stiffnesses in equation (12) in terms of these equivalent constants gives the following condition.

$$\frac{\nu_{eq} E_{eq}}{(1+\nu_{eq})(1-2\nu_{eq})} = \frac{E_{eq}}{2(1+\nu_{eq})} \quad (14)$$

Solving equation (14) for  $\nu_{eq}$  gives the result that  $\nu_{eq}$  is equal to 1/4. Therefore, any uniform three-dimensional space-filling truss structure which is globally isotropic must have an equivalent Poisson's ratio equal to 1/4, and thus, has only one remaining independent elastic constant, its equivalent Young's modulus. Using a similar procedure, it can be shown that two-dimensional space-filling trusses which behave isotropically must have an equivalent Poisson's ratio of 1/3.

### Equivalent Stiffness-to-Density Ratio

Stiffness-to-density ratios are commonly used as indicators of the efficiency of materials. Likewise, equivalent stiffness-to-density ratios are useful indicators of the efficiency of uniform trusses. Most equivalent truss stiffness-to-density ratios are dependent on the design of the truss. However, it will be shown that there exists an equivalent stiffness-to-density ratio which is only a function of the modulus-to-density ratio of the parent material.

In equation (15), a sum of equivalent continuum stiffnesses for a truss is shown to be equal to the sum of the uniaxial stiffnesses of its individual groups of members. Notice that the direction cosine terms drop out because the sum of the squares of the three direction cosines for any member is equal to one.

$$\begin{aligned}
c_{11} + c_{22} + c_{33} + 2c_{23} + 2c_{13} + 2c_{12} &= C_{1111} + C_{2222} + C_{3333} + 2C_{2233} + 2C_{1133} + 2C_{1122} \\
&= \sum_n (C'_{1111})_n (\phi_1^4 + \phi_2^4 + \phi_3^4 + 2\phi_2^2 \phi_3^2 + 2\phi_1^2 \phi_3^2 + 2\phi_1^2 \phi_2^2)_n \\
&= \sum_n (C'_{1111})_n (\phi_1^2 + \phi_2^2 + \phi_3^2)_n^2 \\
&= \sum_n (C'_{1111})_n
\end{aligned} \tag{15}$$

The equivalent density of a space-filling truss is determined by multiplying the density of the parent material,  $\rho$ , by the sum of the volume fractions of all groups of parallel members. Considering equation (7), this relationship can be written as

$$\rho_{eq} = \rho \sum_n v_n = \frac{\rho}{E} \sum_n (C'_{1111})_n \tag{16}$$

Dividing equation (15) by equation (16) gives the following equivalent stiffness-to-density ratio.

$$\frac{c_{11} + c_{22} + c_{33} + 2c_{23} + 2c_{13} + 2c_{12}}{\rho_{eq}} = \frac{E}{\rho} \tag{17}$$

Equation (17) is a unique relationship because it provides a direct correlation between an equivalent continuum stiffness-to-density ratio of the truss, and the modulus-to-density ratio of the parent material in the truss members. Once the parent material is defined for a truss, equation (17) provides a direct relationship between the equivalent anisotropic stiffness of a truss and its equivalent density. This relationship can be used in a number of ways. For example, changes in continuum stiffnesses due to stiffness tailoring of the truss members can be directly translated into a proportional change in equivalent density of the truss. Similarly, requiring the sum of the continuum stiffnesses in the numerator of equation (17) to be constant during stiffness tailoring results in the equivalent density remaining constant. This gives a convenient method for studying the effects of material redistribution within a truss lattice.

Equation (17) can be simplified for trusses that are tailored to be globally isotropic. Without presentation of the details, it can be shown that by writing the equivalent continuum stiffnesses in terms of an equivalent Young's modulus and Poisson's ratio (equal to  $1/4$ ), equation (17) reduces to

$$\frac{E_{eq}}{\rho_{eq}} = \frac{1}{6} \frac{E}{\rho} \tag{18}$$

The significance of equation (18) is that all uniform space-filling trusses which are globally isotropic must have the same equivalent modulus-to-density ratio regardless of their geometries or member sizes. Furthermore, this modulus-to-density ratio must be exactly one sixth of the modulus-to-density ratio of the parent material.

### Equivalent Continuum Strength Tensor

The continuum strength of a truss structure is defined herein as the maximum continuum stress which it can sustain before any of its members buckle elastically. This failure mode is a local phenomenon within the truss lattice which would have one of two effects on the continuum behavior of the truss. If

redundant members exist and load is redistributed, local buckling will cause a change in the continuum stiffnesses of the truss. However, if no load redistribution takes place, local buckling will precipitate a catastrophic failure of the truss lattice. These continuum effects are analogous, respectively, to yielding and ultimate failure in a material.

Since the local failure mode in trusses can be determined analytically it is possible to construct a purely analytical failure theory for trusses. In this section, a tensor which describes the strength of a truss will be constructed, and failure analysis using this strength tensor will be discussed. Having a tensor which represents the strength of a truss is advantageous, because it allows strength to be readily determined in alternate reference frames, or under multiaxial stress states. For materials it is known that strength is not a tensor quantity, and thus, analysis of failure in materials under multiaxial stress can be accomplished only with approximate, semi-empirical theories such as that proposed by von Mises (ref. 14).

The construction of a strength tensor for trusses is based on the assumption that applied stresses can be converted into strains using the compliance equations given in equation (19), and that these strains can be analyzed to determine if the axial compression strain in any truss member has exceeded its critical elastic buckling limit.

$$\begin{pmatrix} \epsilon_{11} \\ \epsilon_{22} \\ \epsilon_{33} \\ 2\epsilon_{23} \\ 2\epsilon_{13} \\ 2\epsilon_{12} \end{pmatrix} = \begin{bmatrix} s_{11} & s_{12} & s_{13} & s_{14} & s_{15} & s_{16} \\ s_{12} & s_{22} & s_{23} & s_{24} & s_{25} & s_{26} \\ s_{13} & s_{23} & s_{33} & s_{34} & s_{35} & s_{36} \\ s_{14} & s_{24} & s_{34} & s_{44} & s_{45} & s_{46} \\ s_{15} & s_{25} & s_{35} & s_{45} & s_{55} & s_{56} \\ s_{16} & s_{26} & s_{36} & s_{46} & s_{56} & s_{66} \end{bmatrix} \begin{pmatrix} \sigma_{11} \\ \sigma_{22} \\ \sigma_{33} \\ \sigma_{23} \\ \sigma_{13} \\ \sigma_{12} \end{pmatrix} \quad (19a)$$

$$\epsilon_{ij} = S_{ijkl} \sigma_{kl} \quad (19b)$$

Note that the compliance matrix in equation (19a) is simply the inverse of the stiffness matrix given in equation (1a). Therefore, the equivalent continuum compliances for a truss can be determined from the equivalent continuum stiffnesses derived previously.

The continuum strains, defined in tensor form in equation (19b), can be transformed into a new coordinate system described by the linear transformation tensor  $T_{ij}$ . The resulting transformed strains,  $\epsilon'_{ij}$ , are

$$\epsilon'_{ij} = T_{io} T_{jp} \epsilon_{op} = T_{io} T_{jp} S_{opkl} \sigma_{kl} \quad (20)$$

The axial strain in any member of the truss is determined by defining an alternate coordinate system with one of its axes aligned along the longitudinal direction of the member, and evaluating the normal strain along that axis. Assuming that the  $I$  axis of the alternate coordinate system is aligned in such a way, the axial strain in the member is given as

$$\epsilon'_{11} = T_{1i} T_{1j} S_{ijkl} \sigma_{kl} = \phi_i \phi_j S_{ijkl} \sigma_{kl} \quad (21)$$

where, as defined before,  $\phi_i$  is the  $i$ th direction cosine of the member.

Failure occurs in a member if its axial strain exceeds a critical value determined for elastic buckling. For the present study, it is assumed that the truss members are slender and therefore buckle as Euler columns (ref. 15), thus the critical strain for the  $n$ th group of members is defined as



$$(\epsilon_{crit.})_n = -\pi^2 \left( \frac{r_n}{l_n} \right)^2 \quad (22)$$

where  $r_n$  is the radius of gyration and  $l_n$  is the length of the members in the  $n$ th group. The minus sign in equation (22) indicates that the critical strain is compressive. A fail-safe criterion can be constructed from equations (21) and (22) by requiring the axial strains in all members to be less than the critical value. This fail-safe criterion can be written as

$$\left[ \frac{(\phi_i \phi_j)_n S_{ijkl}}{-\pi^2 \left( \frac{r_n}{l_n} \right)^2} \right] \sigma_{kl} = [\Omega_{kl}]_n \sigma_{kl} \leq 1 \quad (23)$$

The bracketed term in equation (23) can either be thought of as a third-order tensor representing the strength of the truss, or as a collection of second-order tensors, each representing the strength of a group of parallel members within the truss. The product of this strength tensor and the second-order applied stress tensor,  $\sigma_{kl}$ , is a vector of constants, one for each of the groups of parallel members. For elastic failure to occur, any one of these constants must be greater than or equal to one. Thus the critical stress at which failure occurs is that for which one or more of these constants is equal to one.

Equation (23) represents a purely analytical failure theory for space-filling trusses which can be used with equal ease to analyze strength under multiaxial or uniaxial loading. Similarly, strength in alternate coordinate systems can be readily handled by simply transforming the collection of second-order strength tensors,  $\Omega_{kl}$ , in the same way that a stress or strain tensor would be transformed.

Equation (23) can be used, as described, to determine the strength of a given truss design. Additionally, it is useful for tailoring the strength of a truss design because it is an explicit relationship between the strength of individual members (i.e.,  $r_n/l_n$ ) and the continuum strength of the truss. Strength tailoring is accomplished by varying the strength of individual members to effect a desired change in the continuum strength of the truss. It is important to note that since the continuum compliances of the truss appear in equation (23), strength tailoring is not independent of stiffness tailoring. Consequently, tailoring the continuum stiffnesses of a truss will also change its continuum strength characteristics.

In the remaining sections of this paper, examples of stiffness and strength tailoring of uniform trusses are presented. Truss geometries are selected for analytical simplicity, thus allowing emphasis to be placed on developing an understanding of the analysis techniques.

## EXAMPLES OF STIFFNESS AND STRENGTH TAILORING IN TRUSSES

Equations (9), (17), and (23) provide the basis for analysis of the continuum stiffness, density, and strength of uniform space-filling truss structures. By providing explicit relationships between these continuum quantities and truss design parameters, these equations are effective tools which enable efficient tailoring of the truss stiffness and strength characteristics. In this section, these equations are applied to the analysis of two commonly used truss geometries, and to the tailoring of designs which have continuum isotropic and orthotropic behaviors.

### Regular Octahedral Truss

The Octahedral truss (also known as the Tetrahedral truss, ref. 2, or the Octet truss) is a common geometry that derives its name from the fact that its members connect to form octahedrons and tetrahedrons. For the present study, a regular Octahedral truss is considered which has all identical

members. A repeating cell from this truss is shown in figure 3. The cell contains a regular octahedron at its center (figure 3(a)) and tetrahedrons connected to each of the eight faces of the octahedron (figure 3(b)). Space is filled by translational replication of this cell in each of the three coordinate directions.

Since all members are identical, the octahedral truss has digonal symmetry axes along the lines  $x=y$ ,  $x=z$ , and  $y=z$ ; trigonal symmetry axes along the lines  $x=y=z$ ,  $-x=y=z$ ,  $x=-y=z$ , and  $x=y=-z$ ; and quadragonal symmetry axes along the  $x$ ,  $y$ , and  $z$  axes. This combination of symmetry axes indicates that the regular Octahedral truss is a member of rotational symmetry group  $j$ .

**Calculation of Continuum Stiffness and Density** From table I, it can be seen that the behavior of the regular Octahedral truss is characterized by the three independent elastic constants  $c_{11}$ ,  $c_{12}$ , and  $c_{66}$ . Equation (12) further reduces this number to two. However, these constants lack the relationship  $c_{66}=(c_{11}-c_{12})/2$ ; thus the regular Octahedral truss is not globally isotropic. Values for the elastic constants can be determined from equations (7) and (9). There are six different groups of parallel members in the Octahedral truss, and all members are identical and assumed to have a cross-sectional area of  $A$ . With the half-height of the regular octahedron defined to be  $L$ , as shown in figure 3, the length of each of the members is  $\sqrt{2}L$ . Then, the equivalent unidirectional stiffness for each of the six groups of parallel members is

$$(C'_{1111})_n = \frac{EA}{\sqrt{2} L^2} \quad (24)$$

where  $E$  is the Young's modulus of the material in the members. Substituting equation (24) into equation (9) along with the appropriate direction cosines for the different member groups, gives the result presented in equation (25) for the equivalent continuum stiffness matrix of the Octahedral truss.

$$[c_{mn}] = \frac{EA}{2\sqrt{2} L^2} \begin{bmatrix} 2 & 1 & 1 & 0 & 0 & 0 \\ 1 & 2 & 1 & 0 & 0 & 0 \\ 1 & 1 & 2 & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \quad (25)$$

Notice that the continuum stiffnesses obey the restrictions in table I and equation (12).

Because all members in the regular Octahedral truss are identical, the relative magnitudes of the continuum stiffnesses for the octahedral truss are constrained by the proportions given in the matrix of equation (25). Therefore, changing the axial stiffness of the truss members can only uniformly change all continuum stiffnesses.

The equivalent density of the Octahedral truss can be calculated by substituting the stiffnesses from equation (25) into equation (17). Rearranging and simplifying gives

$$\rho_{eq} = \frac{3\sqrt{2}\rho A}{L^2} \quad (26)$$

**Calculation of Continuum Strength** Before applying equation (23) to calculate the continuum strength of the Octahedral truss, it is necessary to determine the tensor form of the continuum compliances from the stiffness matrix given in equation (25). This is done by inverting the stiffness matrix to get the compliance matrix, and then employing the usual conversion from matrix form to tensor form on the individual compliances (ref. 10). The only remaining unknown truss parameter is the radius of gyration of its members.

Suppose that the strength of the Octahedral truss under a continuum uniaxial compression is required. Assuming this stress to have magnitude  $\sigma_{ult}$  and to be applied along a vector given by the spherical coordinates  $\theta$  and  $\varphi$  (as shown in figure 4), the applied continuum stress tensor can be written as

$$[\sigma_{kl}] = -\sigma_{ult} \begin{bmatrix} (\sin^2\theta \cos^2\varphi) & (\sin^2\theta \sin\varphi \cos\varphi) & (\sin\theta \cos\theta \cos\varphi) \\ (\sin^2\theta \sin\varphi \cos\varphi) & (\sin^2\theta \sin^2\varphi) & (\sin\theta \cos\theta \sin\varphi) \\ (\sin\theta \cos\theta \cos\varphi) & (\sin\theta \cos\theta \sin\varphi) & (\cos^2\theta) \end{bmatrix} \quad (27)$$

The compression strength,  $\sigma_{ult}$ , can be determined for any set of  $\theta$  and  $\varphi$  by substituting equation (27) into equation (23), along with the values for the continuum compliances, the member radius of gyration ( $r$ ), the member length ( $l = \sqrt{2}L$ ), and the appropriate direction cosines. After simplification, equation (23) reduces to a set of six scalar equations ( $n = 1$  to 6), one for each group of parallel members in the truss. Each of these equations can be solved for the value of  $\sigma_{ult}$  that is necessary to cause Euler buckling in the corresponding member. The minimum value determined from these six equations is the lowest uniaxial compression stress at which local buckling occurs within the truss lattice. The lowest value of compression stress found is then defined as the uniaxial compression strength for the given set of  $\theta$  and  $\varphi$ .

A three dimensional plot of the uniaxial compression strength of the Octahedral truss is presented in figure 4 for a range of  $\theta$  and  $\varphi$  from  $0^\circ$  to  $90^\circ$ . Due to symmetry, the strength in all other quadrants is identical. There is a factor of two variation in the compression strength of the lattice and, not surprisingly, the directions of minimum strength are coincident with the directions of the members of the truss. Maximum strength occurs for loading along the three coordinate axes and along the line  $x=y=z$ . The value of the minimum strength is

$$\sigma_{ult} = \frac{EA\pi^2 r^2}{2\sqrt{2} L^4} \quad (28)$$

Since all members are identical, changing the strength of the members would change the vertical scale of the strength plot given in figure 4, but would not change its shape. Introducing member-specific properties will alter the equivalent continuum stiffness and strength; however, this would destroy the geometric symmetry of the lattice and introduce additional independent stiffnesses. In the following section, a truss based on the octahedral lattice is designed for isotropic stiffness, and nearly isotropic strength.

### Isotropic Warren Truss

The lattice of the regular Octahedral truss is modified by adding members that connect all six vertices of each octahedron to the geometric center of the octahedron as shown in figure 5(a). The resulting arrangement of new members forms a cubic lattice within the octahedral lattice, with the edges of the cube lying parallel to the three coordinate axes and each cube containing a regular tetrahedron as shown in figure 5(b). The members of the cubic lattice are of length  $L$  whereas the members of the original octahedral lattice are of length  $\sqrt{2} L$ . This truss geometry is often referred to as the Warren truss because its lattice arrangement is similar to that of a common two-dimensional truss of the same name. Similar to the regular Octahedral truss, the Warren truss is a member of symmetry group  $j$ , and has two independent elastic constants,  $c_{11}$  and  $c_{12}$ . However, unlike the Octahedral truss, the Warren truss has two different members whose relative stiffnesses and strengths can be tailored to affect the continuum behavior of the truss without violating its geometric and elastic symmetry. In this section, it is demonstrated that by redistributing material within the truss lattice the continuum strength and stiffness properties of the lattice can be tailored. In this case, material is transferred from the octahedral lattice

members to the cubic lattice members so that the continuum stiffnesses become isotropic. Also the relative strengths of the members are tailored to reduce variations in continuum compression strength.

**Continuum Stiffness Tailoring** The Warren truss is composed of nine different groups of parallel members. Three groups correspond to the cubic lattice, and six groups correspond to the octahedral lattice. The continuum stiffnesses for the Warren truss can be determined by adding the contributions due to the cubic lattice members to the result presented in equation (25) for the octahedral lattice. The cross-sectional areas of the members in the cubic lattice and the octahedral lattice are defined to be  $A_c$  and  $A_o$ , respectively. Thus, the equivalent uniaxial stiffnesses of the three groups of parallel cubic lattice members are given by

$$(C'_{1111})_n = \frac{EA_c}{L^2} \quad (29)$$

Substituting equation (29) into equation (9) along with the appropriate direction cosines, and adding the result to that presented in equation (25), gives

$$[c_{mn}] = \frac{EA_o}{2\sqrt{2} L^2} \begin{bmatrix} 2 + 2\sqrt{2}\delta_c & 1 & 1 & 0 & 0 & 0 \\ 1 & 2 + 2\sqrt{2}\delta_c & 1 & 0 & 0 & 0 \\ 1 & 1 & 2 + 2\sqrt{2}\delta_c & 0 & 0 & 0 \\ 0 & 0 & 0 & 1 & 0 & 0 \\ 0 & 0 & 0 & 0 & 1 & 0 \\ 0 & 0 & 0 & 0 & 0 & 1 \end{bmatrix} \quad (30)$$

where  $\delta_c$  is defined as  $A_c/A_o$ . It can be seen that if  $\delta_c$  is equal to zero, the cross-sectional area of the cubic lattice members is zero and equation (30) is identical to equation (25). As before, an equivalent density can be calculated using equation (17) and the stiffnesses presented in equation (30). The result is

$$\rho_{eq} = \frac{(3\sqrt{2} + 3\delta_c) \rho A_o}{L^2} \quad (31)$$

To study the effects of redistribution of material within the truss, it is necessary to insure that the total amount of material remains constant. For convenience, the density of the Warren truss is required to be the same as that of the regular Octahedral truss by setting equation (26) equal to equation (31). The result is

$$A_o = \frac{A}{(1 + \delta_c \sqrt{2})} \quad (32)$$

where  $A$  is the cross-sectional area of the members in the regular Octahedral truss analyzed previously. Equation (32) defines the relation between the cross-sectional areas of the cubic and octahedral lattice members within the Warren truss which must hold to keep the equivalent density of the Warren truss equal to that of the regular Octahedral truss. Substituting equation (32) into equation (30) gives explicit equations for the continuum stiffnesses of the Warren truss in terms of the member area ratio,  $\delta_c$ . To better understand the effects of redistribution of material, the stiffness components in equation (30) are translated into equivalent Young's modulus, Poisson's ratio, and shear modulus as follows.

$$E_{eq} = \frac{(c_{11} + 2c_{12})(c_{11} - c_{12})}{c_{11} + c_{12}} = \frac{4EA(1 + 2\sqrt{2}\delta_c)}{2\sqrt{2} L^2 (3 + 2\sqrt{2}\delta_c)} \quad (33)$$

$$\nu_{eq} = \frac{c_{12}}{c_{11} + c_{12}} = \frac{1}{3 + 2\sqrt{2}\delta_c} \quad (34)$$

$$G_{eq} = c_{66} = \frac{EA}{2\sqrt{2} L^2 (1 + \delta_c \sqrt{2})} \quad (35)$$

These stiffness components are plotted in figure 6 as functions of the area ratio,  $\delta_c$ . For  $\delta_c$  equal to zero, no material has been redistributed from the octahedral lattice to the cubic lattice, and the stiffnesses represent those of the Octahedral truss. As  $\delta_c$  is increased, material is moved from the octahedral lattice to the cubic lattice, and this is accompanied by an increase in the equivalent Young's modulus and decreases in the equivalent Poisson's ratio and the equivalent shear modulus. From equations (34) and (35) it can be seen that as  $\delta_c$  becomes large, both the Poisson's ratio and the shear modulus approach zero. This effect is consistent with the fact that the cubic lattice of members is not a kinematically stable truss by itself. Because of this, it is unreasonable to consider designs having very large values of  $\delta_c$ .

For the Warren truss to be globally isotropic, it is necessary for its stiffnesses to satisfy the following condition.

$$G_{eq} = \frac{E_{eq}}{2(1 + \nu_{eq})} \quad (36)$$

Substituting the expressions from equations (33) - (35) into equation (36) shows that  $\delta_c$  must be equal to  $1/(2\sqrt{2})$  for isotropy. Substituting this value of  $\delta_c$  into equation (32) gives a value of  $4A/5$  for the cross-sectional area of the members in the octahedral lattice, and consequently a value of  $\sqrt{2}A/5$  for the cross-sectional area of the members in the cubic lattice. Thus, if one fifth of the material that was originally in the members of the octahedral truss is redistributed into the members of the cubic lattice, the resulting truss behaves isotropically. The isotropic values for the equivalent Young's modulus, Poisson's ratio, and shear modulus are

$$(E_{eq})_{iso.} = \frac{EA}{\sqrt{2} L^2}, \quad (\nu_{eq})_{iso.} = \frac{1}{4}, \quad (G_{eq})_{iso.} = \frac{\sqrt{2}EA}{5 L^2} \quad (37)$$

Notice that the equivalent isotropic Poisson's ratio is  $1/4$  which is the value that was predicted earlier for globally isotropic trusses. Also, calculating the ratio of the equivalent isotropic Young's modulus (equation (37)) to the equivalent density (equation (26)) gives the result predicted in equation (18) for globally isotropic trusses.

**Continuum Strength Tailoring** Applying the same procedure used for the Octahedral truss, it is possible to determine the continuum strength of the isotropic Warren truss and to evaluate the effects on continuum strength of varying the strength of the truss members. For the purpose of comparison, the same continuum stress tensor given in equation (27) is also applied to the Warren truss. Two cases are analyzed. In the first, it is assumed that all members in the truss have the same radius of gyration, and in the second it is assumed that all members have the same buckling load. The first case is representative of a truss with thin-walled members of equal cross-sectional diameter. The second case illustrates the effects of tailoring individual member buckling strengths on the continuum strength of the truss.

For the first case, the radius of gyration of all members is equal to  $r$ , and the lengths of the members are  $L$  for the cubic lattice, and  $\sqrt{2}L$  for the octahedral lattice. These values are substituted into equation (23) along with the continuum compliances determined from equation (30) and the appropriate direction cosines. The result is a set of nine scalar equations, one for each group of parallel members in the truss, from which the minimum value of  $\sigma_{ult}$  is determined for the given set of  $\theta$  and  $\phi$ .

A three dimensional plot of the uniaxial compression strength of the isotropic Warren truss is presented in figure 7 for the same range of  $\theta$  and  $\phi$  as before. The shape of the strength plot is very similar to that of the Octahedral truss, and despite the redistribution of material from the octahedral lattice, the values and the directions of the minimum and maximum strength are the same as those for the Octahedral truss. The directions and maximum strength are coincident with the directions of the cubic lattice members, and the directions of minimum strength are coincident with the directions of the octahedral lattice members. Selecting all members to have the same radius of gyration causes the cubic lattice members to have twice the buckling load of the octahedral lattice members, because of the difference in their lengths. This effect causes the factor of two variations in the continuum strength.

Variation in truss strength might not be a concern for many design applications, however if it is desirable to have a truss which behaves isotropically in stiffness, it is likely that it is also desirable for it to be isotropic in strength. By tailoring the buckling loads of the cubic lattice members to be the same as those of the octahedral lattice, the variations in continuum strength can be significantly reduced. For this case, the radius of gyration of the cubic lattice members is reduced to  $r/\sqrt{2}$  so that the buckling loads of all members are the same. A plot of the resulting continuum compression strength is presented in figure 8. Although some variation still exists in the continuum strength, the magnitude of the variation has been significantly reduced.

The use of three dimensional strength plots is particularly helpful for developing strength tailoring rules, because they provide visualization of the correlation between member orientations and continuum strength variations. Without this correlation it would be difficult to develop strength tailoring rationale for the members. The example presented is fairly simple due to the isotropic stiffness behavior and geometric symmetry of the Warren truss. Therefore, the correlation between variations in continuum strength and the orientation of members is fairly obvious. However, for trusses with less geometric symmetry or more complex applied stress tensors, this correlation might not be apparent without the use of a three dimensional strength plot.

### Orthotropic Warren Truss

Many applications exist for large truss structures with orthotropic, rather than isotropic, continuum properties. For orthotropic applications the requirements on continuum stiffness and strength are much higher in one direction than others. For example, many applications involve beam-like trusses which primarily carry bending and/or torsional loads. In these cases, the longitudinal (along the beam's length) stiffness and strength requirements are much higher than the transverse stiffness and strength requirements. Therefore, it is probably more efficient to use a truss with orthotropic continuum properties than one with isotropic properties.

From table I it can be seen that trusses of symmetry groups  $i$  and  $j$  are not candidates for orthotropic design because their stiffnesses (and strengths) must be the same in all three coordinate directions. Trusses of all other symmetry groups are candidates for orthotropic tailoring because their properties in the  $z$  direction can differ from those in either the  $x$  or the  $y$  directions. The truss presented in figure 9 is a variation of the Warren truss design which is a member of symmetry group  $f$ , and is thus a likely candidate for orthotropic design. The lattice arrangement of this truss is identical to that of the Warren truss except the length of the repeating cell in the  $z$  direction differs from that in either the  $x$  or the  $y$  directions by the proportion  $\beta$ . The purpose of this section is to apply stiffness and strength tailoring techniques to generate orthotropic designs which have high stiffnesses and strengths in the  $z$  direction but have the same equivalent density as that of the isotropic Warren truss.

**Calculation of Continuum Stiffnesses** The orthotropic Warren truss shown in figure 9 has four different members. The cross-sectional areas for members of groups 1 and 2 are defined as  $\delta_1 A$  and  $\delta_2 A$ , respectively, where  $\delta_1$  and  $\delta_2$  are variable area ratios, and  $A$  is the cross-sectional area assumed

earlier for the members in the Octahedral truss. The equivalent uniaxial stiffnesses for groups of these members are determined using equation (7) and the results are given in equations (38) and (39).

$$(C'_{IIII})_1 = \frac{\delta_1 EA}{L^2} \quad (38)$$

$$(C'_{IIII})_2 = \frac{\delta_2 EA(1 + \beta^2)^{1/2}}{2\beta L^2} \quad (39)$$

For simplicity, it is assumed that members of groups 3 and 4 are the same as those in the isotropic Warren truss. Therefore, the cross-sectional area of members of group 3 is  $\sqrt{2}A/5$ , and the cross-sectional area of members of group 4 is  $4A/5$ . The equivalent uniaxial stiffnesses are the same for member groups 1 and 2 and the value of this stiffness is given in equation (40).

$$(C'_{IIII})_3 = (C'_{IIII})_4 = \frac{\sqrt{2}EA}{5\beta L^2} \quad (40)$$

Substituting these uniaxial stiffnesses and the appropriate transformation tensors into equation (9) and simplifying gives the following values for the non-zero continuum stiffnesses.

$$c_{11} = c_{22} = \frac{EA}{\beta L^2} \left[ \frac{2\sqrt{2}}{5} + \frac{\delta_2}{(1 + \beta^2)^{3/2}} \right] \quad (41)$$

$$c_{12} = c_{66} = \frac{\sqrt{2}EA}{5\beta L^2} \quad (42)$$

$$c_{13} = c_{23} = c_{44} = c_{55} = \frac{EA}{\beta L^2} \left[ \frac{\beta^2 \delta_2}{(1 + \beta^2)^{3/2}} \right] \quad (43)$$

$$c_{33} = \frac{EA}{\beta L^2} \left[ \delta_1 \beta + \frac{2\beta^4 \delta_2}{(1 + \beta^2)^{3/2}} \right] \quad (44)$$

Notice that these stiffnesses obey the conditions presented in table I and equations (12) and (13) for trusses of symmetry group  $f$ . Equations (41) through (44) are explicit functions of the three remaining design parameters,  $\beta$ ,  $\delta_1$ , and  $\delta_2$ . Therefore, these equations can be used directly to determine how variations in the design parameters affect the orthotropic characteristics of the truss.

An equivalent density can be calculated for the orthotropic Warren truss by substituting the stiffnesses from equations (41) through (44) into equation (17). The result is

$$\rho_{eq} = \frac{\rho A}{\beta L^2} \left[ \frac{6\sqrt{2}}{5} + \delta_1 \beta + 2(1 + \beta^2)^{1/2} \delta_2 \right] \quad (45)$$

Setting equation (45) equal to equation (26) insures that the equivalent density of the orthotropic Warren truss is the same as that of the regular Octahedral truss and the isotropic Warren truss. The resulting expression can be rearranged to give the following condition on the area ratio  $\delta_2$ .

$$\delta_2 = \frac{(3\sqrt{2} - \delta_1)\beta - 6\sqrt{2}/5}{2(1 + \beta^2)^{1/2}} \quad (46)$$

Equation (46) reduces the set of independent design parameters to the repeating cell length ratio,  $\beta$ , and the cross-sectional area ratio,  $\delta_1$ .

An equivalent z-direction Young's modulus can be determined for the orthotropic Warren truss by inverting the  $s_{33}$  component of the compliance matrix as follows.

$$(E_{eq})_z = \frac{1}{s_{33}} \quad (47)$$

Performing this calculation gives the result

$$(E_{eq})_z = \frac{\sqrt{2}EA [15\delta_1\sqrt{2} + 18\beta^3 - 5(\delta_1\sqrt{2} - 6\beta/5)^2]}{L^2 [15 - 5\delta_1\sqrt{2} + 12\beta + 6\beta^3]} \quad (48)$$

To determine the improvement in stiffness in the z direction, the modulus given in equation (48) is divided by the Young's modulus of the isotropic Warren truss given in equation (37). The resulting normalized z-direction Young's modulus is

$$\frac{(E_{eq})_z}{(E_{eq})_{iso.}} = \frac{30\delta_1\sqrt{2} + 36\beta^3 - 10(\delta_1\sqrt{2} - 6\beta/5)^2}{15 - 5\delta_1\sqrt{2} + 12\beta + 6\beta^3} \quad (49)$$

A three-dimensional plot of the normalized z-direction Young's modulus is presented in figure 10 for ranges of the length ratio,  $\beta$ , and the cross-sectional area ratio,  $\delta_1$ . The isotropic Warren truss is characterized by  $\delta_1 = \sqrt{2}/5$  and  $\beta = 1$ , this point on the plot corresponds to a normalized z modulus equal to 1. As  $\delta_1$  is increased, for a fixed value of  $\beta$ , material is transferred from members of group 2 to members of group 1 (see figure 9). This causes an increase in the z modulus because the group 1 members are oriented parallel to the z direction. As  $\beta$  is increased, for a fixed value of  $\delta_1$ , the number of group 3 and group 4 members in a given volume decreases. Thus, to maintain constant density, material is redistributed among group 1 and group 2 members also causing an increase in the z modulus.

**Calculation of Continuum z-Direction Strength** The strength of the orthotropic Warren truss is calculated for a uniform continuum compression applied in the z-direction. This applied stress tensor is given in equation (50), and is substituted into equation (23).

$$[\sigma_{kl}] = \begin{bmatrix} 0 & 0 & 0 \\ 0 & 0 & 0 \\ 0 & 0 & -(\sigma_{ult})_z \end{bmatrix} \quad (50)$$

Due to their alignment parallel to the z direction, members in group 1 buckle at lower continuum stresses than the remaining members in the truss (this result was verified through additional analysis not presented herein). Thus, considering only buckling in group 1 members, equation (23) can be reduced to equation (51), where  $r_l$  and  $l_l$  are the radius of gyration and length of members in group 1.



$$(\sigma_{ult})_z = \frac{\pi^2 r_I^2}{l_I^2 s_{33}} \quad (51)$$

Defining the radius of gyration of these members to be  $r$  and their length to be  $\beta L$  (see figure 9), and substituting the result from equation (47) gives the following expression for the z-direction compression strength of the orthotropic Warren truss.

$$(\sigma_{ult})_z = \frac{\pi^2 r^2}{\beta^2 L^2} (E_{eq})_z \quad (52)$$

The z-direction compression strength of the isotropic Warren truss can be determined from figure 7 ( $\theta = 0^\circ$ ), and this value can be used to normalize equation (52). The result is

$$\frac{(\sigma_{ult})_z}{(\sigma_{ult})_{iso.}} = \frac{(E_{eq})_z}{\beta^2 (E_{eq})_{iso.}} \quad (53)$$

Unlike the z modulus, the factor of  $\beta^2$  in the denominator of equation (53) causes the z-direction strength to decrease with increasing  $\beta$ . However, it should be apparent that both modulus and strength have the same variation with the cross-sectional area ratio,  $\delta_I$ . A three-dimensional plot of the normalized z-direction compression strength is presented in figure 11 for comparison to the modulus plot in figure 10. Since both modulus and strength increase as  $\delta_I$  is increased, it is best to select the largest practical value for  $\delta_I$ . As an example, if the cross-sectional areas of all members within the truss are constrained such that they differ by no more than a factor of five, the maximum allowable value for  $\delta_I$  would be  $\sqrt{2}$ . Assuming this value for  $\delta_I$  gives the following for all of the member cross-sectional areas.

$$A_1 = \sqrt{2}A \quad , \quad A_2 = \frac{(10\beta - 6)A}{5(2 + 2\beta^2)^{1/2}} \quad , \quad A_3 = \sqrt{2}A/5 \quad , \quad A_4 = 4A/5 \quad (54)$$

A plot of the normalized z-direction strength and modulus is presented in figure 12 assuming  $\delta_I$  is equal to  $\sqrt{2}$ . As already explained, extending the length of the Warren truss cell in the z direction (increasing  $\beta$ ) increases the stiffness while decreasing the strength of the truss. Therefore, the optimum length for the truss cell depends on the relative importance of continuum strength and continuum stiffness in the design.

## CONCLUDING REMARKS

A deterministic procedure has been presented for tailoring the continuum stiffness and strength of uniform space-filling truss structures through the appropriate selection of truss geometry and member sizes (i.e., flexural and axial stiffnesses and length). A key aspect of this procedure is symbolic manipulation of the equivalent continuum constitutive equations to produce explicit relationships between truss member sizes and continuum strength and stiffness. To aid in the selection of an appropriate truss geometry for a given application, a finite set of possible geometric symmetry groups which can be possessed by uniform trusses was presented, and the implied elastic symmetry associated with each geometric symmetry group was identified.

Equivalent continuum stiffness were determined using an existing technique which assumes that the displacement field within a truss is single-valued, and the members within a truss carry only axial load.

Based on these assumptions, it was shown that generally anisotropic trusses are characterized by 18 independent elastic constants rather than 21 as is normal for a generally anisotropic solid. It was also shown that this result guarantees that all three-dimensional trusses which behave isotropically, in a continuum sense, must have an equivalent Poisson's ratio of  $1/4$ . Furthermore, a direct relationship was derived between an anisotropic stiffness-to-density ratio of a truss and the stiffness-to-density ratio of its parent material. Using this relationship it was shown that the equivalent Young's modulus-to-density ratio of any isotropic three-dimensional truss is exactly  $1/6$  times the modulus-to-density ratio of the parent material of the truss.

A purely analytical failure theory was developed for trusses by defining failure to be elastic buckling of any member within the truss lattice. This theory allows the construction of a strength tensor which simplifies failure analysis under multiaxial stress and alternate coordinate systems.

To illustrate the application of these analysis techniques, truss designs were developed which behave isotropically and orthotropically under continuum loading. In these examples, stiffness tailoring was accomplished through redistribution of material among the truss members, and strength tailoring was accomplished by varying the relative buckling strengths of the members. This deterministic approach to the analysis and tailoring of truss behavior can significantly enhance the understanding of relationships between the design parameters and the continuum elastic behavior of trusses. Ultimately, this improved understanding should enable the creation of more efficient truss designs.

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TABLE I.- ELASTIC CHARACTERISTICS OF ROTATIONAL SYMMETRY GROUPS

| Symmetry Group <sup>a</sup> | Conditions on stiffnesses                                                                                                                                                                                            | Independent Constants |
|-----------------------------|----------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------------|-----------------------|
| no symmetry                 | none                                                                                                                                                                                                                 | 21                    |
| <i>a</i>                    | $c_{14}, c_{15}, c_{24}, c_{25}, c_{34}, c_{35}, c_{46}, c_{56} = 0$                                                                                                                                                 | 13                    |
| <i>b</i>                    | same as <i>a</i> along with $c_{16}, c_{26}, c_{36}, c_{45} = 0$                                                                                                                                                     | 9                     |
| <i>c</i>                    | $c_{16}, c_{26}, c_{34}, c_{35}, c_{36}, c_{45} = 0$ , $c_{11} = c_{22}$ , $c_{44} = c_{55}$ ,<br>$c_{13} = c_{23}$ , $c_{14} = -c_{24} = c_{56}$ , $c_{15} = -c_{25} = -c_{46}$ ,<br>$c_{66} = (c_{11} - c_{12})/2$ | 7                     |
| <i>d</i>                    | same as <i>c</i> along with $c_{15}, c_{25}, c_{46} = 0$                                                                                                                                                             | 6                     |
| <i>e</i>                    | same as <i>a</i> along with $c_{36}, c_{45} = 0$ ,<br>$c_{11} = c_{22}$ , $c_{44} = c_{55}$ , $c_{13} = c_{23}$ , $c_{16} = -c_{26}$                                                                                 | 7                     |
| <i>f</i>                    | same as <i>e</i> along with $c_{16}, c_{26} = 0$                                                                                                                                                                     | 6                     |
| <i>g</i>                    | same as <i>c</i> along with $c_{14}, c_{15}, c_{24}, c_{25}, c_{46}, c_{56} = 0$                                                                                                                                     | 5                     |
| <i>h</i>                    | same as <i>g</i>                                                                                                                                                                                                     | 5                     |
| <i>i</i>                    | same as <i>b</i> along with $c_{11} = c_{22} = c_{33}$ ,<br>$c_{12} = c_{13} = c_{23}$ , $c_{44} = c_{55} = c_{66}$                                                                                                  | 3                     |
| <i>j</i>                    | same as <i>i</i>                                                                                                                                                                                                     | 3                     |

<sup>a</sup>See figure 2.

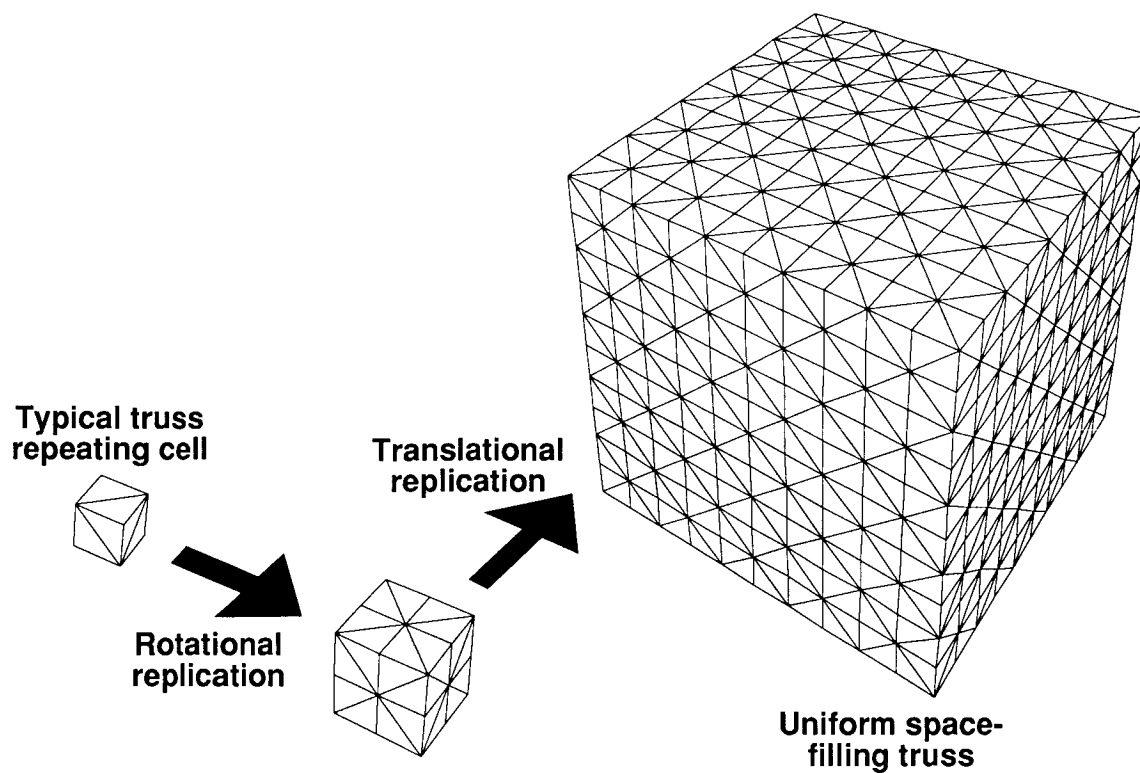


Figure 1. Large uniform trusses are generated from a repeating cell.

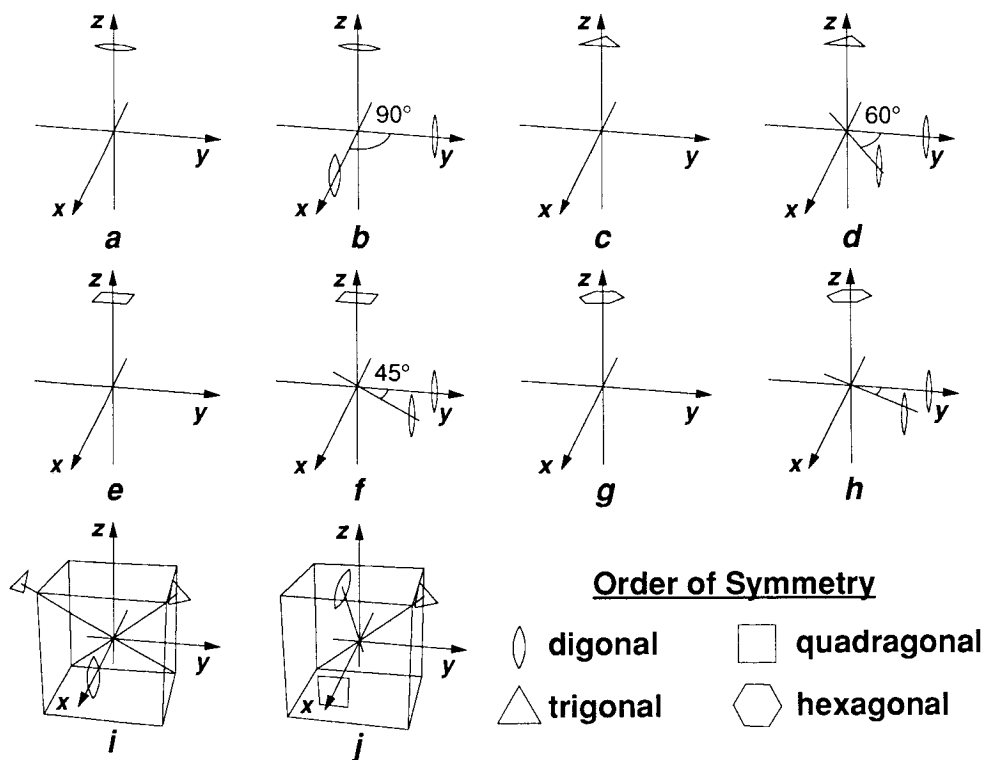
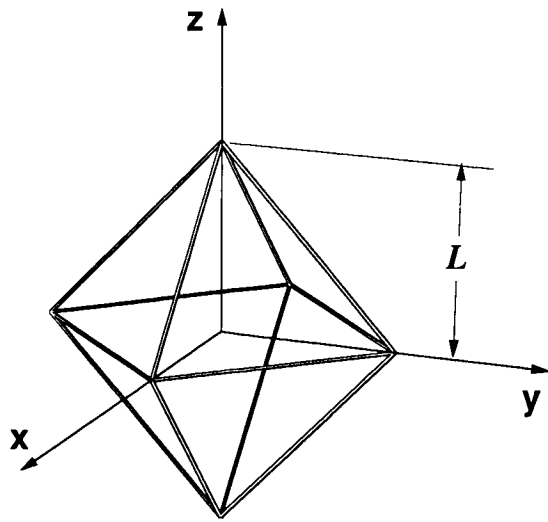
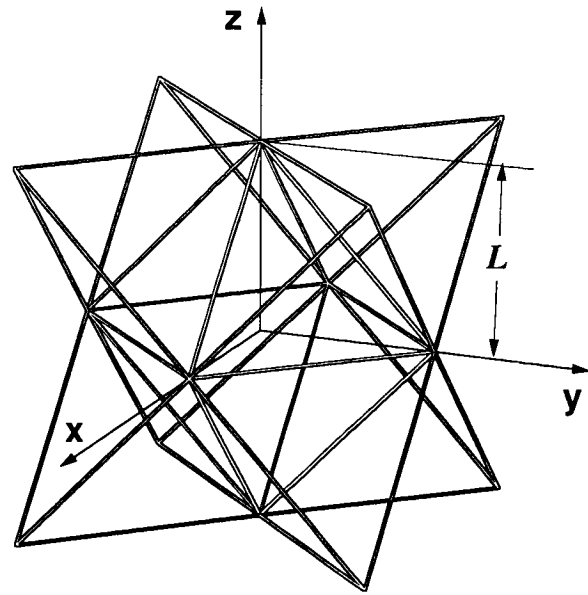


Figure 2. Possible rotational symmetry groups.

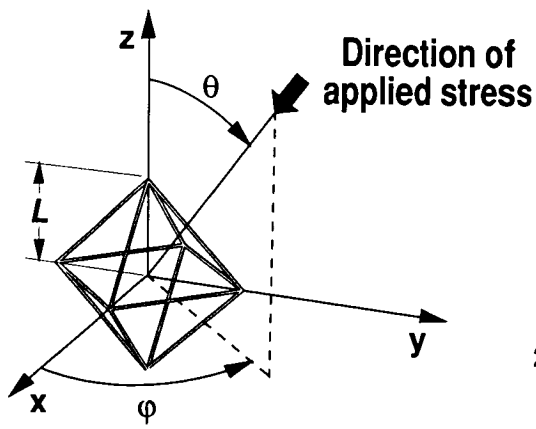


(a) regular octahedron.



(a) complete repeating cell with regular tetrahedrons.

Figure 3. Repeating cell for regular Octahedral truss.



$$\left( \frac{\sigma_{ult}}{\frac{EA\pi^2 r^2}{2\sqrt{2}L^4}} \right)$$

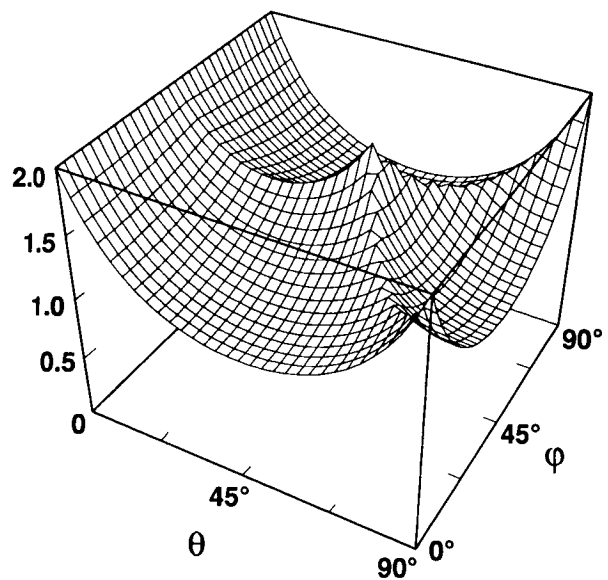
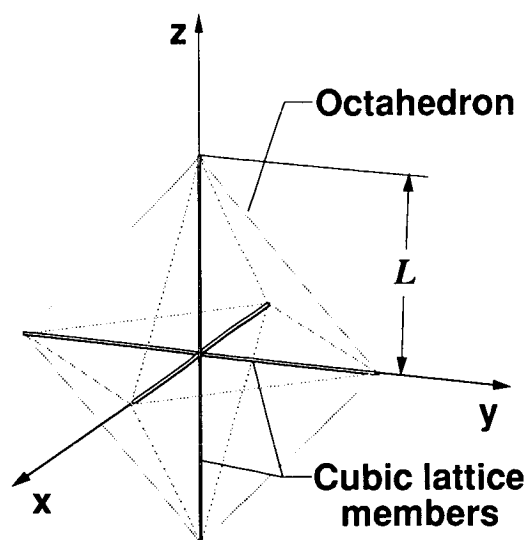
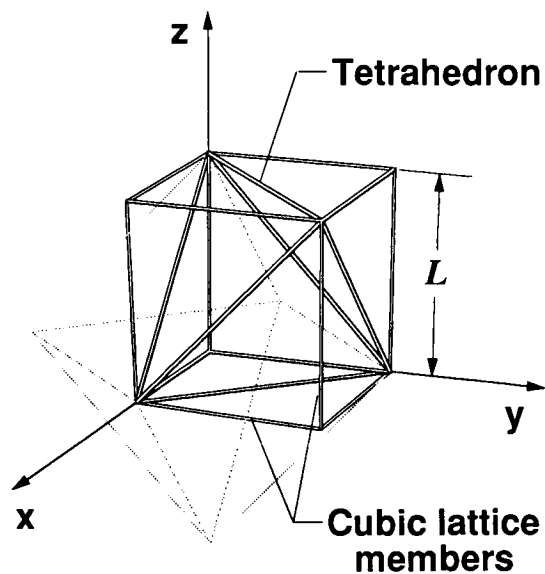


Figure 4. Strength of Octahedral truss under uniaxial compression.



(a) members added to octahedral lattice.



(b) resulting cubic lattice.

Figure 5. Repeating cell for Warren truss.

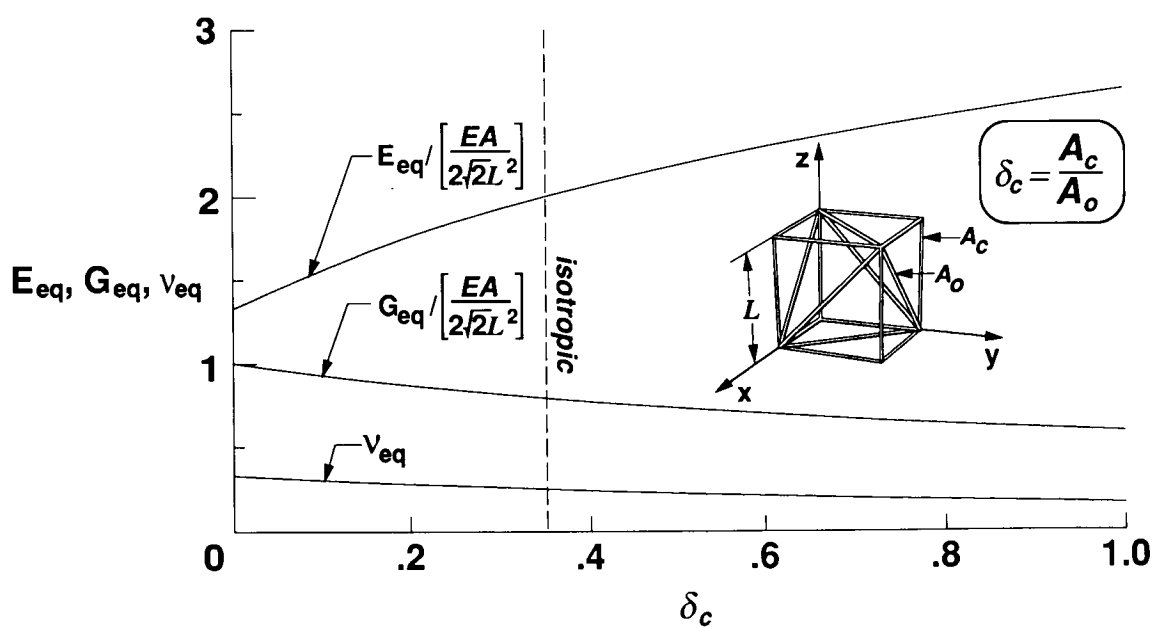


Figure 6. Stiffness tailoring of the Warren truss.

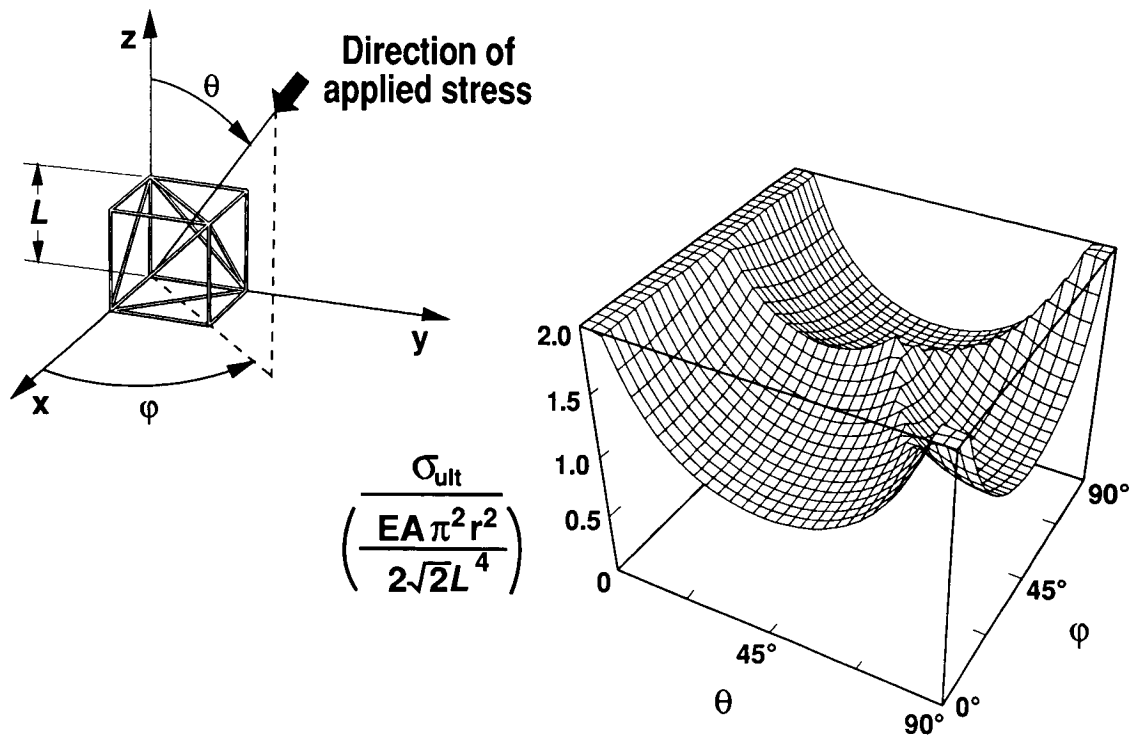


Figure 7. Uniaxial compression strength of isotropic Warren truss (all members have same radius of gyration).

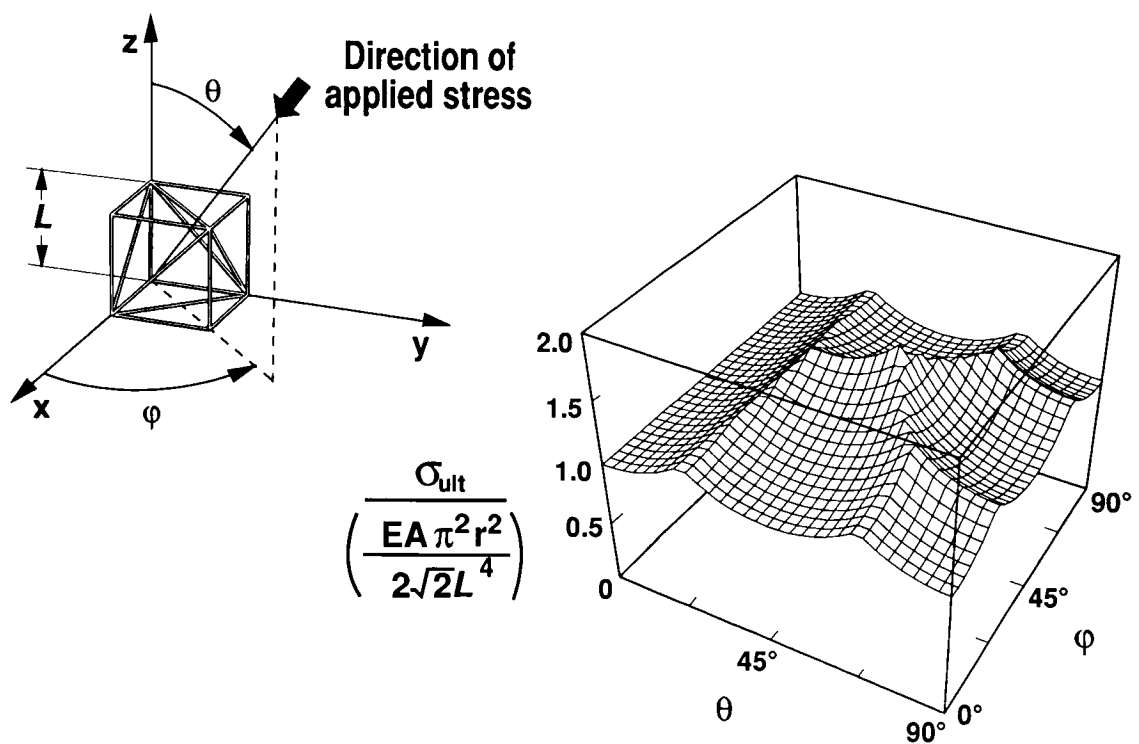


Figure 8. Variation in strength diminished by tailoring all members to have the same buckling load.



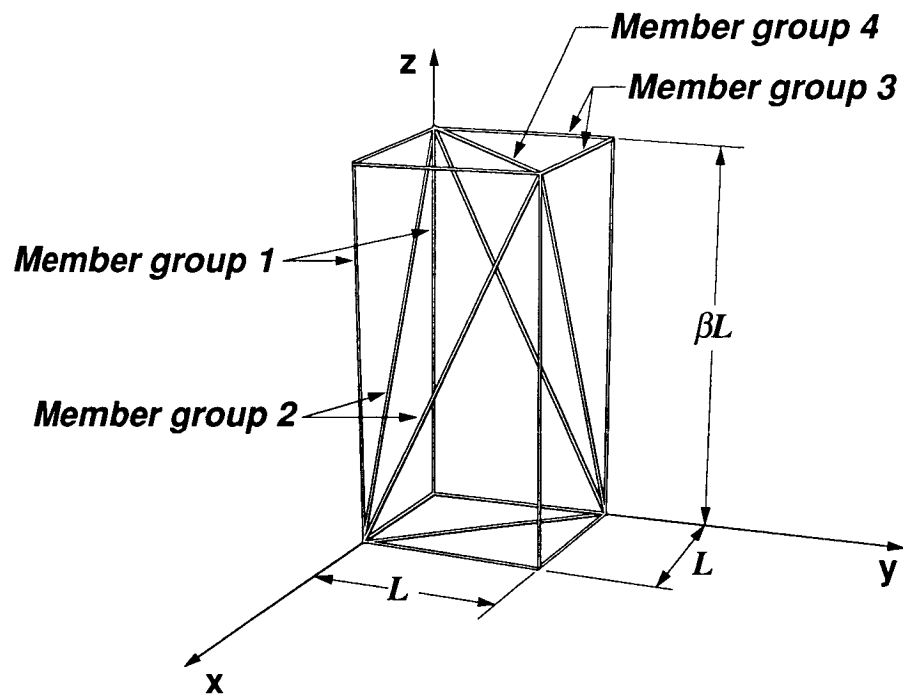


Figure 9. Repeating cell for orthotropic Warren truss.

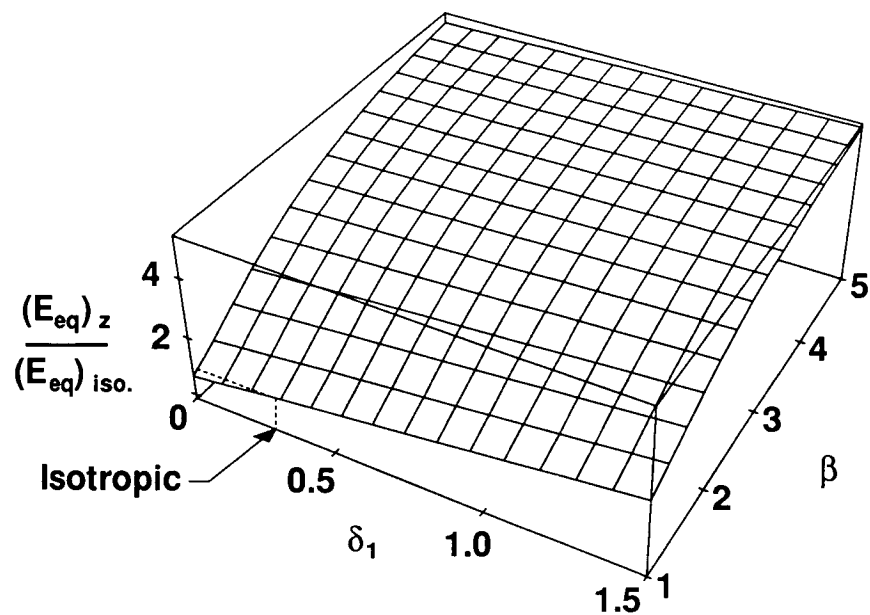


Figure 10. Variation in z modulus with cell length ratio,  $\beta$  and member area ratio,  $\delta_1$ .

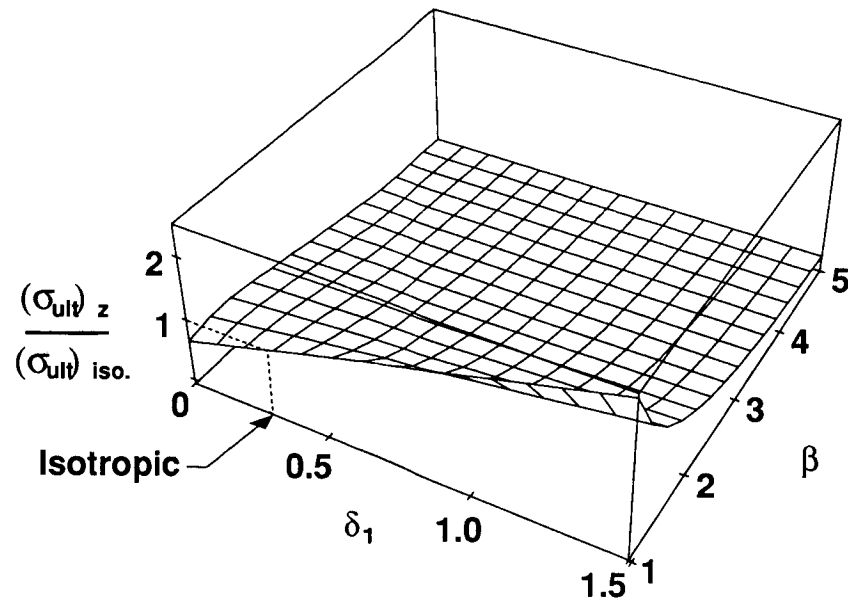


Figure 11. Variation in z strength with cell length ratio,  $\beta$  and member area ratio,  $\delta_1$ .

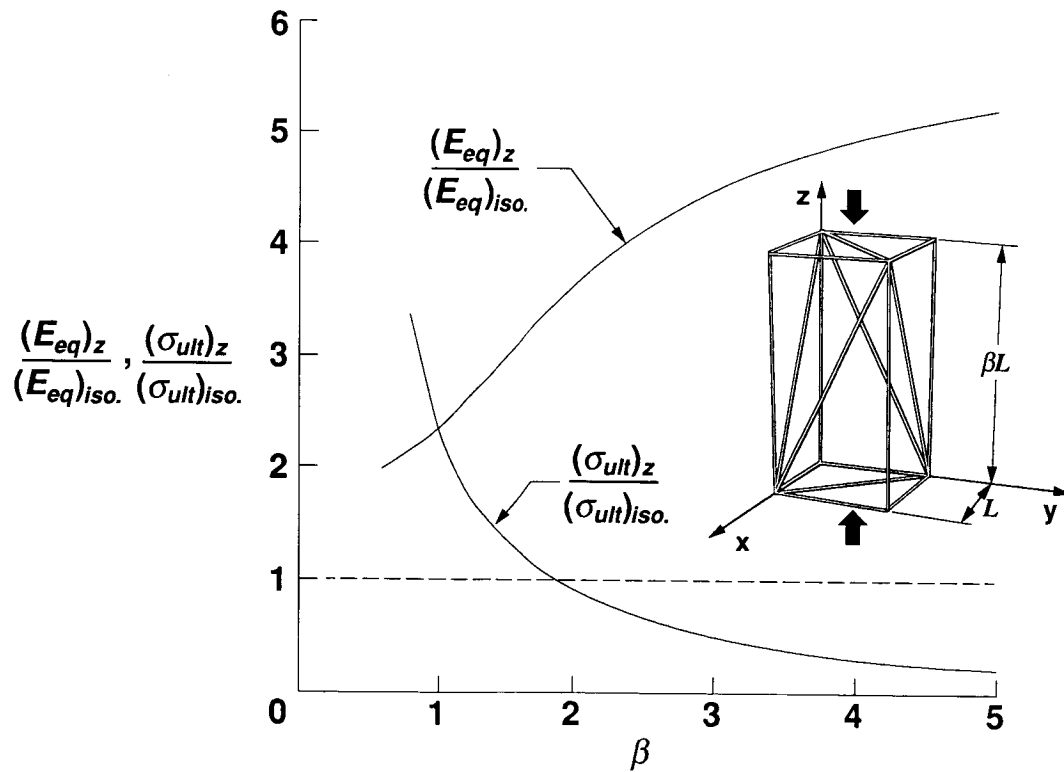


Figure 12. Variation in z direction stiffness and strength with cell length.

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DESIGN PREDICTION FOR LONG TERM STRESS RUPTURE SERVICE  
OF COMPOSITE PRESSURE VESSELS<sup>†</sup>

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SUMMARY

Extensive stress rupture studies on glass composites and Kevlar composites were conducted by the Lawrence Radiation Laboratory beginning in the late 60s and extending to about 8 years in some cases. Some of the data from these studies published over the years were incomplete or were tainted by spurious failures, such as grip slippage. These data have been carefully resurveyed by cognizant staff. Updated verified data sets have been defined for both fiberglass and Kevlar composite strand test specimens. These updated data are analyzed in this report by a convenient form of the bivariate Weibull distribution, to establish a consistent set of design prediction charts that may be used as a conservative basis for predicting the stress rupture life of composite pressure vessels.

The updated glass composite data exhibit an invariant Weibull modulus with lifetime. The data are analyzed in terms of homologous service load (referenced to the observed median strength). The equations relating life, homologous load, and probability are given, and corresponding design prediction charts are presented. A similar approach is taken for Kevlar composites, where the updated strand data do show a turndown tendency at long life accompanied by a corresponding change (increase) of the Weibull modulus. This turndown characteristic is not present in stress rupture test data of Kevlar pressure vessels. A modification of the stress rupture equations is presented to incorporate a latent, but limited, strength drop, and design prediction charts are presented that incorporate such behavior.

The methods presented utilize Cartesian plots of the probability distributions (which are a more natural display for the design engineer), based on median normalized data that are independent of statistical parameters and are readily defined for any set of test data. A technique is shown for estimating the Weibull modulus from each observed value. The design prediction equations and the corresponding design prediction charts can be set up to provide selected levels of conservatism in those regions where data are sparse or unavailable. Design values based on these single-end data should be conservative for multiple-end roving and massive composite structures like those on pressure vessels.

<sup>†</sup> This work was presented at the Ninth DoD/NASA/FAA Conference on Fibrous Composites in Structural Design, 4-7 November, Lake Tahoe, Nevada. Funding for this effort was processed through Contract F04701-88-C-0089 under an interagency agreement from the U.S. Department of the Navy.

## INTRODUCTION

The purpose of this report is to provide a useful engineering tool for estimating stress rupture life of S-glass and Kevlar composites subject to long-term tensile stress. The data base is drawn from the extensive 10 year period test program conducted on S-glass and Kevlar single-end composite strands at the Lawrence Livermore National Laboratory (LLNL).

Previously published results of S-glass exhibited a turndown characteristic at long times. This characteristic, when analyzed by a quadratic maximum likelihood method (Ref. 1), produced unrealistic life prediction extrapolation and did not yield realistic and useful long life predictions. The LLNL S-glass data base has recently been subject to a searching re-examination by the cognizant staff (Refs. 2, 3) to eliminate spurious failures such as grip pullouts. These updated data are analyzed here by the Weibull model and tabulated in Appendix A. These data do not exhibit a turndown and are well approximated by invariant Weibull moduli for the life and strength distributions. It should be noted that the stress rupture analysis will produce straight line probability contours on log stress vs log life coordinates if the Weibull moduli for life and strength are invariant with time. The converse is also true.

Convenient analytical methods based on the Weibull model are presented. These methods allow an estimation of the Weibull modulus based on each ranked observation and allow estimation of the population median life based on partial (censored or incomplete) data. The methods of analysis are simple and straightforward, and may be applied in the same fashion to other stress rupture data. The Weibull modulus for S-glass life is  $b = 0.9$ , based on the updated stress rupture data presented. LLNL measurements of S-glass composite strand strength show scatter with a representative coefficient of variation  $CV = 0.48+$ , corresponding with  $m = 24.9$ , which is used in the tabulated data and design prediction. These parameters for S-glass are used to produce the design chart of Figure 1.

The LLNL Kevlar composite strand data (Ref. 2) are presented in Appendix B. These data show a turndown trend. Kevlar spherical vessel data (Ref. 4) do not show this trend. The Weibull model was modified, as discussed later, to incorporate a strength degradation that develops later in time according to a first-order reaction rate (Ref. 5) with a specific time constant. This model predicts that within the time period where degradation is expected, the Weibull modulus for life should show an apparent increase. That is exactly what the Kevlar strand data show, and this modified form is used to construct a rational design chart that includes a partial degradation ( $f = 0.3$ ) with a time constant  $t_c \approx 120,000$  hours. The Weibull modulus for Kevlar composite strand life is also  $b = 0.9$ , while the Weibull modulus for strength is  $m \approx 30$ , reflecting a typically slightly lower scatter ( $CV = 0.04$ ) for Kevlar composite strands than for the S-glass strands. The resulting design chart is shown in Figure 2. Over-plotted on this chart are the pressure vessel data, which do not exhibit the turndown and show greater life at the same homologous load.

Published data for carbon fiber composite strands are very sparse (Ref. 6) and extend to shorter times than the S-glass and Kevlar data. Figure 3 is a preliminary design chart, constructed using the indicated parameters, and should be useful for first-order life estimates of carbon composite pressure vessel stress rupture life.

## DISCUSSION

### The Bivariate Weibull Distribution

An expression for the stress rupture (Ref. 7) of a multifilament strand is given by

$$S = \text{Exp} -H\{R,t\}$$

where  $H$  is a function of the applied load and time. Assuming the function to be separable and of

exponential (power-law) form, it can be expressed as a bivariate form of the well known Weibull distribution function (Ref. 7).

The two-parameter form of the Weibull distribution is given by

$$S = \text{Exp} - \{X/X_o\}^m$$

As a matter of convenience, the function is normalized here to the median value  $X_m$ , or to any other percentile  $S_r$ ,  $X_r$ , which may be appropriate, as in the case of a partial sample where the median has not yet been reached in the experimental program and is unknown:

$$S = \text{Exp} - \{\ln(1/S_r) (X/X_r)^m\}$$

If the median is known, then  $S_r = 0.5$  and the distribution scale parameter becomes the median. From a design engineer's point of view, this is intuitively more meaningful than the Weibull scale parameter  $X_o$ , which corresponds with the 63% quantile. Forms of the Weibull distribution that use mean normalized data are more awkward because the quantile of the mean value involves a gamma function of the Weibull modulus (shape parameter):

$$S = \text{Exp} - \{\Gamma(1+1/m)(X/X_{avg})\}^m$$

The bivariate Weibull form, normalized to median strength and life, is given by

$$S = \text{Exp} - \{\ln(2) (R_m)^m (t/t_m)^b\}$$

The symbol  $R$  is the homologous load referenced to the median value. This form is used in subsequent discussions for the analysis of the composite stress rupture data.

### From Filaments to Composite Structures

The susceptibility of single filaments to stress rupture is invariably more severe than the susceptibility of multifilament strands for several reasons. The filament surface is totally exposed to the surrounding environment; the access of environmental factors such as moisture, air, or radiation is not impeded; and access remains unimpeded throughout the exposure time. The filament failure under these influences is total, i.e., no load can be transmitted after filament failure. The situation is less severe for the multifilament strand, especially if twist is present. Within a multifilament strand, there is some inhibition of diffusion by geometric effects and by the gradual development of internal concentration gradients. Furthermore, individual filament breaks might not lead to load-carrying reduction because frictional coupling and twist act like the matrix of a composite strand. The composite strand is even less susceptible to individual filament rupture failure because the matrix encapsulates the filaments and can offer considerable protection from diffusion and effects of the environments. Consequently, stress rupture experiments show relatively early failure times and turndown characteristics for single filaments and bare strands as compared with composite strands.

When comparing lifetimes at equal homologous loads, still another factor enters that causes the multifilament strands to exhibit longer life. The prevailing stress within the constituent filaments is reduced because the median strength of a multifilament strand is expected to be less than the median strength of single filaments, and the discrepancy increases as the variability of filament strength increases. At the same homologous load, the average filament stress in massive composites is reduced from the single filament or single-end strand.

For these reasons, it is concluded that using single-end composite strands to establish design charts for massive composites, such as filament-wound pressure vessels, will provide an inherently conservative basis for estimation.

## ANALYTICAL FORMS

### Invariant Weibull Moduli

The median normalized bivariate Weibull equation (Ref. 7) relates load, life, and probability, as shown below:

$$S = \text{Exp} - \left\{ \text{Ln}(2) \left( \frac{\sigma}{\sigma_m} \right)^m \left( \frac{t}{t_m} \right)^b \right\}$$

The equation is the power law form with the underlying linear log-log relation :

$$\left( \frac{\sigma}{\sigma_m} \right)^m \left( \frac{t}{t_m} \right)^b = \frac{\left( \text{Ln} \left( \frac{1}{S} \right) \right)}{\text{Ln}(2)}$$

This relation defines a family of straight lines on log-log coordinates in which the following proportionality holds (where R represents the load fraction of median strength):

$$R^m t^b \propto \frac{\left( \text{Ln} \left( \frac{1}{S} \right) \right)}{\text{Ln}(2)}$$

The preceding relations may be used as shown in the following steps to compute desired combinations of load fraction, R, life, t, and survival probability, S. The shape parameters (Weibull moduli) for strength, m, and for life, b, are determined by analysis of experimental strength and life distributions. For example, the LLNL glass data give  $m = 24.9$  and  $b = 0.9$ . The procedure to construct the probability, load, life design chart follows.

1. Select a reference set of values from the data or design:  $R_r$ ,  $t_r$ , and  $S_r$  (usually  $S_r = 0.5$ ).
2. Select the load and life Weibull moduli  $m$  and  $b$  to give desired conservatism for the design chart.
3. Compute the constant K

$$K = \frac{R_r^m t_r^b \text{Ln}(2)}{\text{Ln} \left( \frac{1}{S_r} \right)}$$

4. Define the general R, t, S relation by

$$R^m t^b = \left[ \frac{K \text{Ln} \left( \frac{1}{S} \right)}{\text{Ln}(2)} \right]$$

5. Construct the design chart using the log-log form ( $S_r = 0.5$ )

$$\text{Log}(R) = \text{Log}(R_r) - \frac{b}{m} \text{Log}\left(\frac{t}{t_r}\right) + \left(\frac{1}{m}\right) \text{Log}\left[\frac{\text{Ln}\left(\frac{1}{S}\right)}{\text{Ln}(2)}\right]$$

This equation, representing time invariant Weibull moduli for both strength and life, is typified by the glass design chart of Figure 1.

6. Compute life for particular load fraction,  $R$ , and survival probability,  $S$ , by

$$t = \left\{ \frac{K}{R^m} \left[ \frac{\text{Ln}(1/S)}{\text{Ln}(2)} \right] \right\}^{(1/b)}$$

7. Compute load fraction for particular life,  $t$ , and survival probability,  $S$ , by

$$R = \left\{ \frac{K}{t^b} \left[ \frac{\text{Ln}(1/S)}{\text{Ln}(2)} \right] \right\}^{(1/m)}$$

8. Compute the survival probability for particular load fraction,  $R$ , and life,  $t$ , by

$$S = \text{Exp} \left\{ -\frac{R^m t^b}{K} \text{Ln}(2) \right\}$$

#### Single Point Estimates for Median Life and Weibull Modulus

The following equations show how the median life may be estimated, when the shape factor,  $m$ , is known, from each early observed failure life at  $t_r$  corresponding with the survival probability,  $S_r$ , where

$S_r = 1 - (r-0.5)/N$  and  $r$  is the rank serial number. The use of  $(r-0.5)$  as the effective rank for computing the corresponding probability is a convenience. Alternative formulations are  $r/N$ ,  $r/(N+1)$ ,  $(r-0.3)/(n+0.4)$ , as well as probabilistic treatments of the "correct" rank assignments. Extensive computer simulations of Weibull distributions (Ref. 8) show that  $(r-0.5)/N$  is an effective and convenient rank assignment. The value of  $S_r$  was taken to be  $1 - (r-0.5)/N$  in the tabulated computations. The median life value is estimated by the following equations (with  $m$  known):

$$S = \text{Exp} \left\{ -\text{Ln}(2) \left( \frac{t}{t_m} \right)^m \right\}$$

$$t_{0.5} = t_m = t_r \left( \frac{\text{Ln}(2)}{\text{Ln}\left(\frac{1}{S_r}\right)} \right)^{\frac{1}{m}}$$



The shape factor may also be estimated from a few early observations, and in this case we take as an interim normalizing value the longest observed life  $t_r$ , corresponding to survivability  $S_r$ , and use the preceding data at lesser life  $t$  having survival probability  $S$  (greater than  $S_r$ ) to estimate the Weibull Modulus by the relation:

$$m = \frac{\text{Ln} \left\{ \frac{\text{Ln} \left( \frac{1}{S} \right)}{\text{Ln} \left( \frac{1}{S_r} \right)} \right\}}{\text{Ln} \left( \frac{t}{t_r} \right)}$$

The two estimators are combined to produce individual median life estimates from each observed value, with  $t_r$  as the normalizing life at survival probability,  $S_r$ , as shown below

$$t_{0.5} = t_r \left\{ \frac{\text{Ln} (2)}{\text{Ln} \left( \frac{1}{S_r} \right)} \right\}^{\left\{ \frac{\text{Ln} \left( \frac{t}{t_r} \right)}{\text{Ln} \left\{ \frac{\text{Ln} \left( \frac{1}{S} \right)}{\text{Ln} \left( \frac{1}{S_r} \right)} \right\}} \right\}}$$

These predictions are tabulated in the data appendices.

#### Degradation and Strength Turndown

The possibility of a latent strength reduction, which may appear after a substantial amount of time under sustained load, was addressed by Christensen (Ref. 5). He showed the effect on stress rupture life of degradation that follows first-order chemical reaction rates, with the strength decreasing progressively until failure:

$$\sigma = \sigma_r \text{Exp} (-t/t_c)$$

This idea, with some additional constraints, can be used to make a simple and useful modification of the design charts. One can postulate such degradation, which might be related to moisture effects, and the breakdown of locally susceptible regions in the load-carrying filaments. These locally susceptible regions are finite; after all of them are degraded, the breakdown process will stop. This leads to a fraction,  $f$ , of strength lost in accordance with the time constant,  $t_c$ , of this assumed reaction

$$\sigma = \sigma_r [1 - f (1 - \text{Exp} (-t/t_c))]$$

This fractional strength loss is incorporated into the stress rupture equations as shown below:

$$R = R_r \left( \frac{t_r}{t} \right)^{\frac{b}{m}} \left[ \frac{\text{Ln} \left( \frac{1}{S} \right)}{\text{Ln} (2)} \right]^{\frac{1}{m}} \left\{ 1 - f \left[ 1 - \text{Exp} \left( -\frac{t}{t_c} \right) \right] \right\}$$

The log-log form to be used for plotting the probability contours on the design chart is given by

$$\text{Log} (R) = \text{Log}(R_r) - \frac{b}{m} \text{Log} \left( \frac{t}{t_r} \right) + \left( \frac{1}{m} \right) \text{Log} \left[ \frac{\text{Ln} \left( \frac{1}{S} \right)}{\text{Ln} (2)} \right] + \text{Log} \left\{ 1 - f \left[ 1 - \text{Exp} \left( -\frac{t}{t_c} \right) \right] \right\}$$

This type of design chart is illustrated by the Kevlar composite strand design chart of Figure 2, although the Kevlar vessel data do not exhibit the turndown.

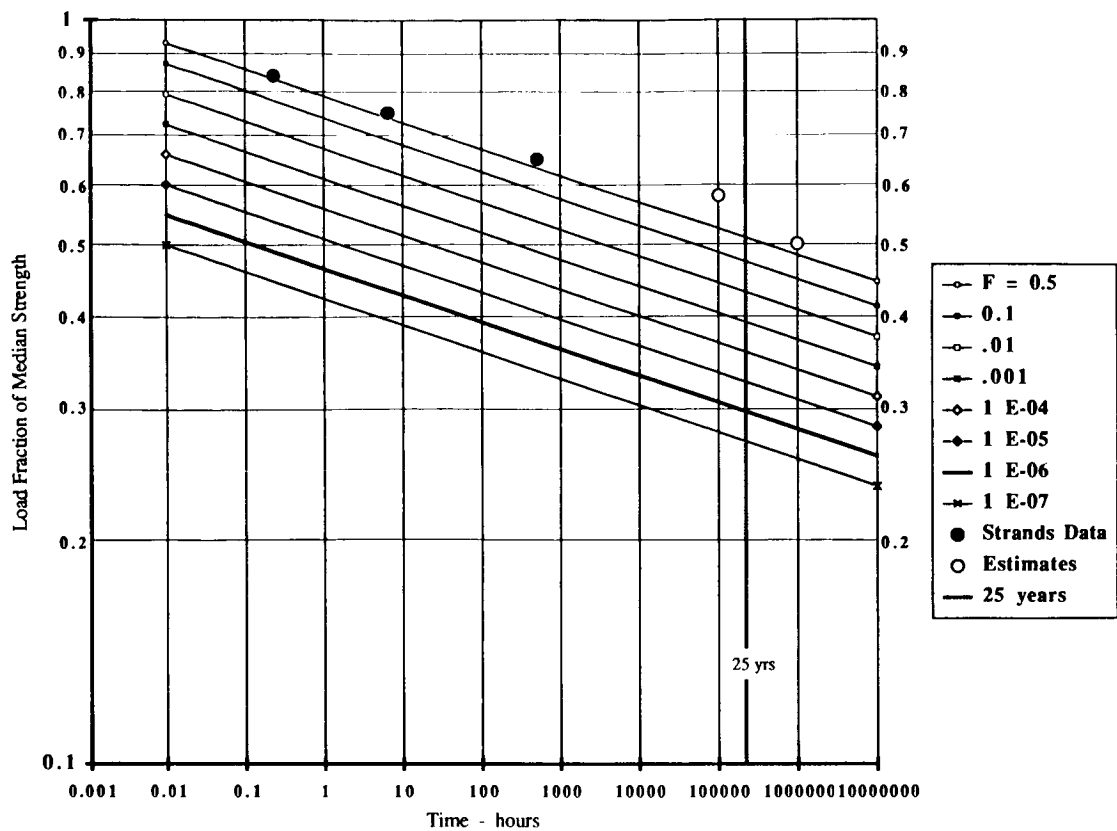


Figure 1. Glass Composite Strand Stress Rupture Design Chart

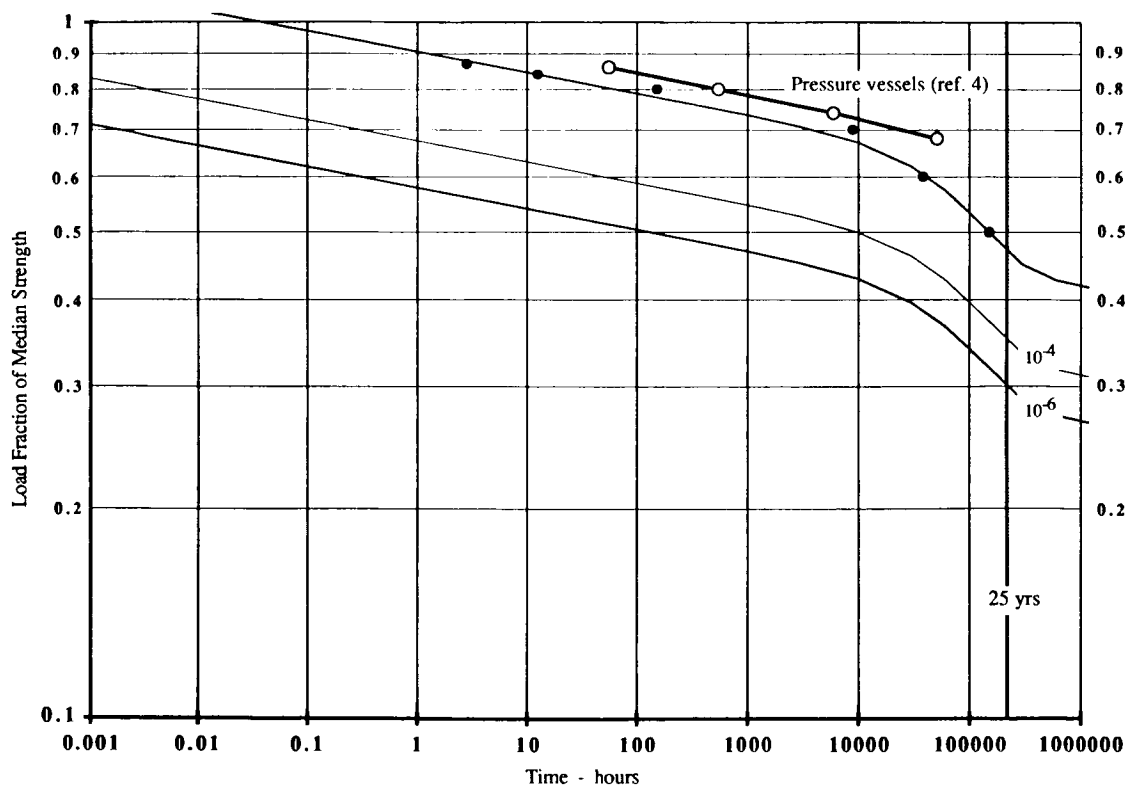


Figure 2. Kevlar Composite Strand Stress Rupture Design Chart  
with  $m = 30$   $b = 0.9$   $f = 0.3$   $T_c = 120000$  hrs.

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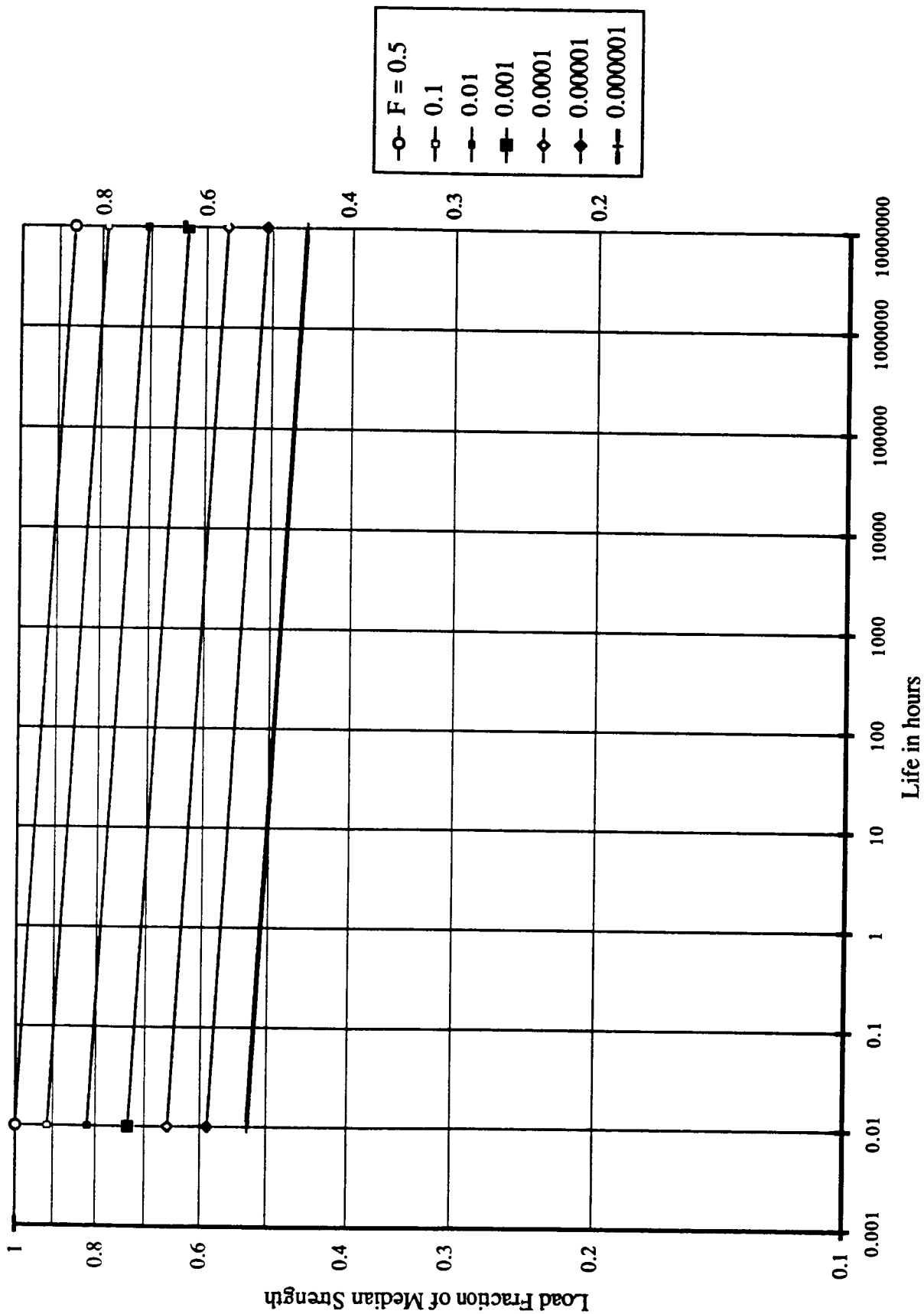


Figure 3. Graphite Composite Stress Rupture Design Chart  
with  $m = 21$   $b = 0.15$

## CONCLUSIONS

Design charts based on the bivariate form of the Weibull distribution are presented for S-glass composites and Kevlar composites, and for carbon composites in preliminary form. The equations are readily modified to accommodate selected values for the life and strength Weibull moduli in order to determine the most consistent and useful fits to the data.

The methods presented may be applied to any type of stress rupture data that exhibit a consistent power law relation between the Weibull moduli for strength and life. A modification of the linear model is presented and used to accommodate the Kevlar strand data turndown tendency at long life.

The underlying analytical models show that if life and strength are Weibull distributions with invariant moduli (shape factors), then log load vs log life will be linear. Conversely, if the log load vs log life is linear, and either the life distribution or the strength distribution is a Weibull, then the other distribution must also be a Weibull.

The data processing by these methods allows each observation to be used for estimating the Weibull modulus, giving a valuable perspective for engineering approximations that seek a single conservative design reference value. Methods for estimating median life from incomplete data are also shown.

A notable point is the dramatic difference seen in stress rupture life between S-glass and Kevlar composite rupture data on the one hand, and the carbon data on the other hand. The carbon data seem to exhibit very little stress rupture degradation, and therefore offer very high homologous stresses in operation. Such high stress potential (and high performance) may not be a practically usable characteristic. The lower design stresses required for the glass and Kevlar also provide a certain amount of damage tolerance during the service life. In addition, both S-glass and Kevlar are inherently resistant to moderate impact and casual damage. Carbon composites, on the other hand, are notorious for susceptibility to physical damage and abuse. Such susceptibility, coupled with very high operating stresses, could lead to premature or catastrophic failures in cases of casual damage to a carbon composite vessel operating so close to its expected strength.

The design of S-glass and Kevlar pressure vessels is controlled by stress rupture characteristics for long-term service. The design of carbon composite pressure vessels for long term service is controlled by the amount and type of damage in the operating environment. The carbon composite pressure vessels must be protected from environmental damage or designed to resist and tolerate the service environment.

## Acknowledgement

The author acknowledges the assistance of the Commercial Products Division of Structural Composites Industries Inc. for opportunities to examine the current stress rupture models and the performance records for large commercial composite vessel production.

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## SYMBOLS

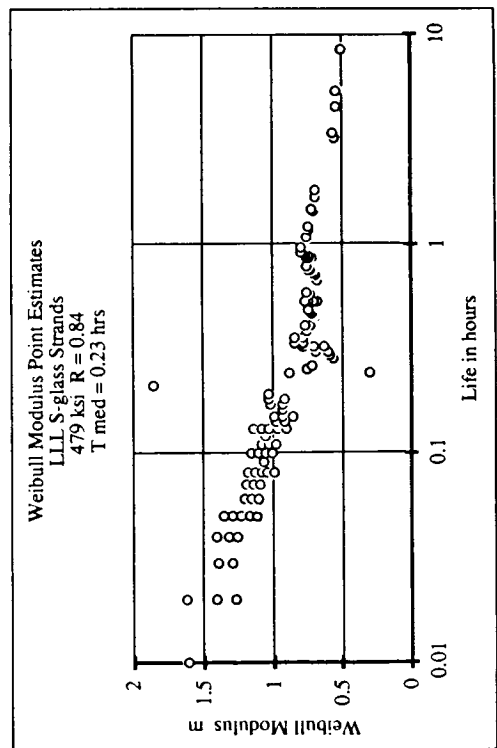
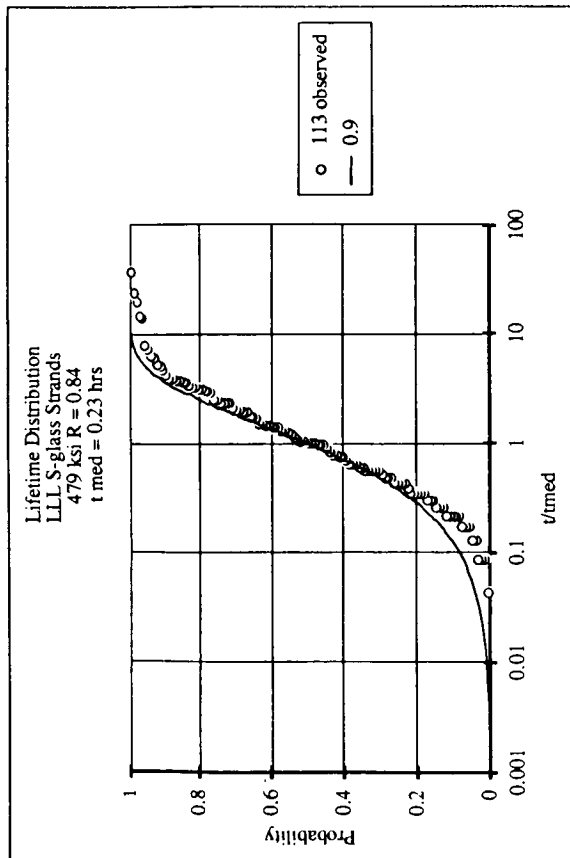
- |                   |                                                                                                        |
|-------------------|--------------------------------------------------------------------------------------------------------|
| S,F               | Cumulative survival and failure probabilities.                                                         |
| m                 | Weibull modul (shape factor) for strength; subscript designates median.                                |
| b                 | Weibull modulus for life.                                                                              |
| R                 | Homologous load (referenced to the median strength).                                                   |
| H, X              | Function of load and time, and generic variable.                                                       |
| t                 | Time, life.                                                                                            |
| r                 | Subscript for reference value of load and time; also rank number for sorted data.                      |
| f, t <sub>c</sub> | Strength fraction lost in first-order degradation reaction, with characteristic time, t <sub>c</sub> . |
| σ                 | Stress or strength.                                                                                    |

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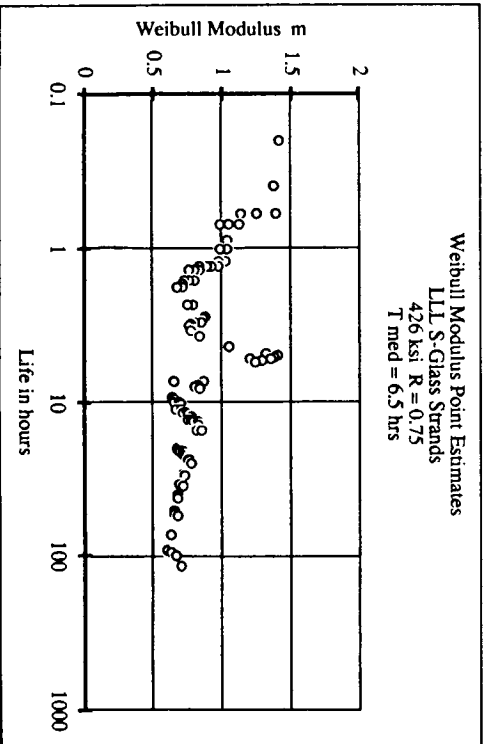
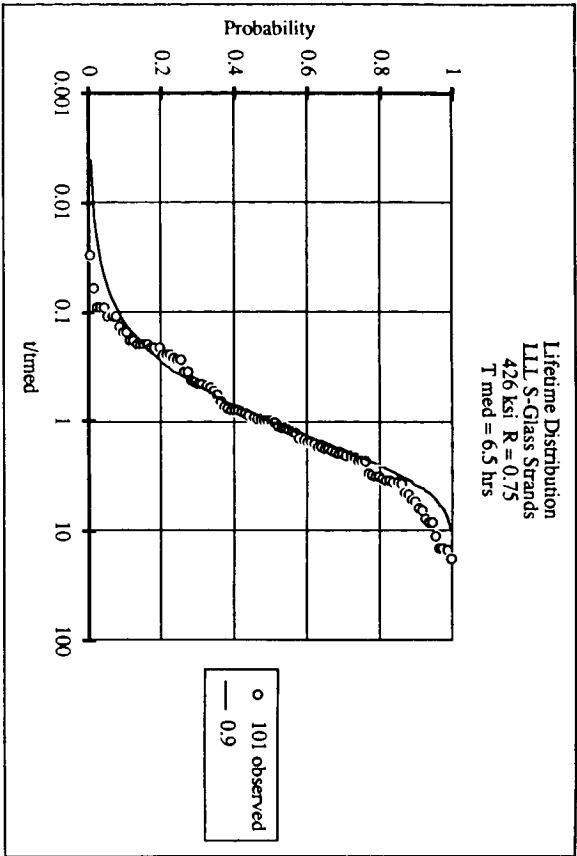
## APPENDIX A

### S-Glass Composite Strand Data Tables and Plots Based on LLNL Data Update

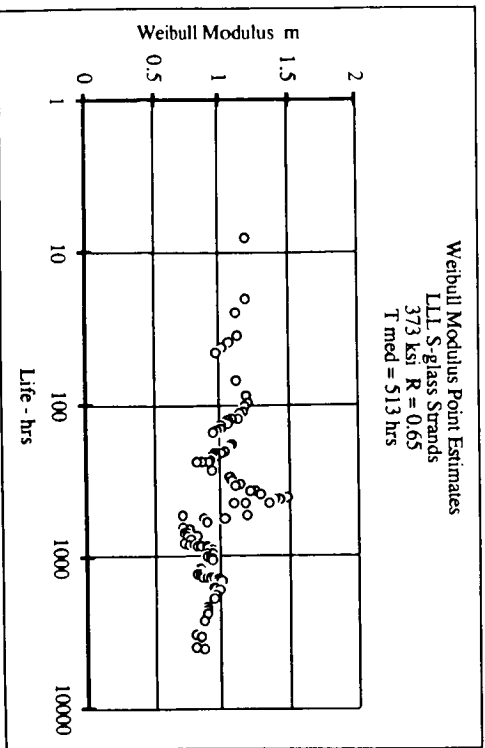
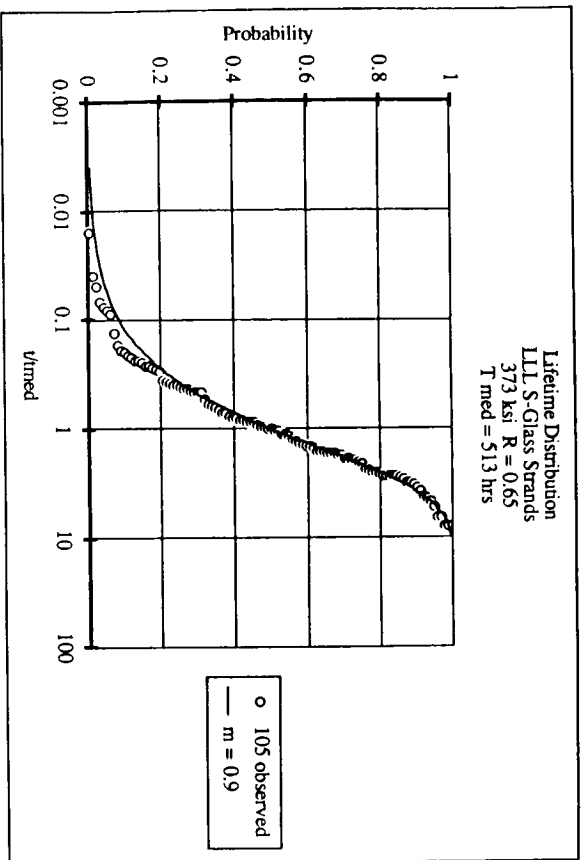




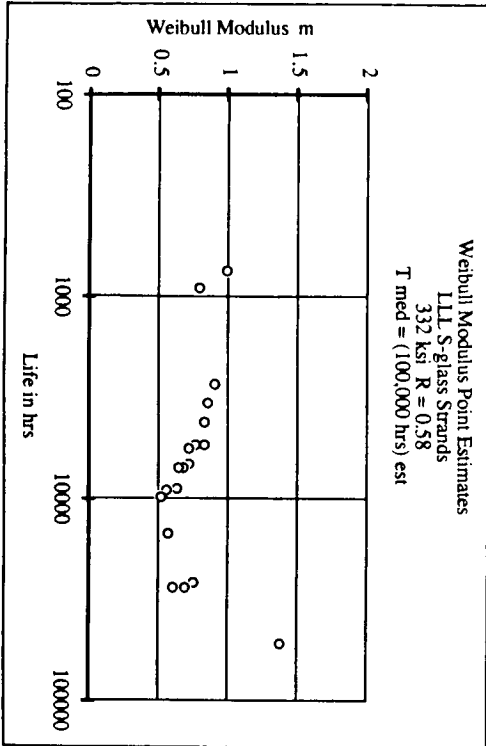
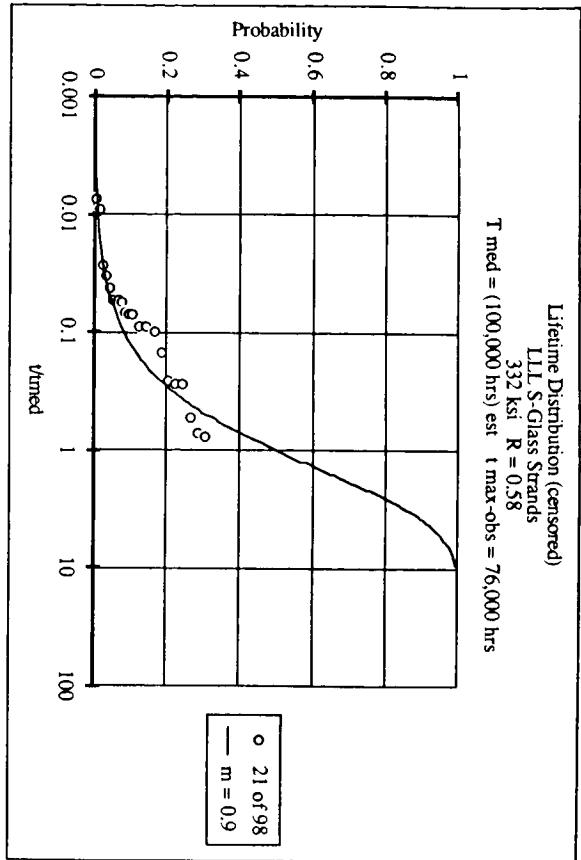
| LLL S-glass   | Rank | Life t<br>hrs | Normal<br>$t/t_{med}$ | Prob<br>F | est<br>b | fit<br>b=0.9 | Rank<br>r | Life t<br>hrs | Normal<br>$t/t_{med}$ | Prob<br>F | est<br>b | fit<br>b=0.9 |
|---------------|------|---------------|-----------------------|-----------|----------|--------------|-----------|---------------|-----------------------|-----------|----------|--------------|
| Stress = 479  | 1    | 0.01          | 0.043                 | 0.004     | 1.6      | 0.004        | 58        | 0.24          | 1.043                 | 0.509     | 0.6      | 1.029        |
|               | 2    | 0.02          | 0.087                 | 0.013     | 1.6      | 0.012        | 59        | 0.24          | 1.043                 | 0.518     | 1.2      | 1.058        |
| $R = 0.84$    | 3    | 0.02          | 0.087                 | 0.022     | 1.4      | 0.022        | 60        | 0.25          | 1.087                 | 0.527     | 0.9      | 1.088        |
|               | 4    | 0.02          | 0.087                 | 0.031     | 1.3      | 0.032        | 61        | 0.26          | 1.130                 | 0.535     | 0.8      | 1.118        |
| $N = 113$     | 5    | 0.03          | 0.130                 | 0.040     | 1.4      | 0.043        | 62        | 0.28          | 1.217                 | 0.544     | 0.6      | 1.150        |
|               | 6    | 0.03          | 0.130                 | 0.049     | 1.3      | 0.054        | 63        | 0.29          | 1.261                 | 0.553     | 0.6      | 1.182        |
| Median = 0.23 | 7    | 0.04          | 0.174                 | 0.058     | 1.4      | 0.065        | 64        | 0.3           | 1.304                 | 0.562     | 0.7      | 1.214        |
|               | 8    | 0.04          | 0.174                 | 0.066     | 1.3      | 0.077        | 65        | 0.32          | 1.391                 | 0.571     | 0.7      | 1.248        |
| Specimen      | 9    | 0.04          | 0.174                 | 0.075     | 1.2      | 0.089        | 66        | 0.32          | 1.391                 | 0.580     | 0.7      | 1.282        |
| type 1        | 10   | 0.05          | 0.217                 | 0.084     | 1.4      | 0.101        | 67        | 0.32          | 1.391                 | 0.588     | 0.7      | 1.317        |
| (572)         | 11   | 0.05          | 0.217                 | 0.093     | 1.3      | 0.113        | 68        | 0.32          | 1.391                 | 0.597     | 0.8      | 1.353        |
| sgl end       | 12   | 0.05          | 0.217                 | 0.102     | 1.2      | 0.126        | 69        | 0.33          | 1.435                 | 0.606     | 0.8      | 1.389        |
|               | 13   | 0.05          | 0.217                 | 0.111     | 1.2      | 0.139        | 70        | 0.33          | 1.435                 | 0.615     | 0.9      | 1.427        |
|               | 14   | 0.05          | 0.217                 | 0.119     | 1.1      | 0.152        | 71        | 0.35          | 1.522                 | 0.624     | 0.8      | 1.466        |
|               | 15   | 0.06          | 0.261                 | 0.128     | 1.2      | 0.166        | 72        | 0.35          | 1.522                 | 0.633     | 0.9      | 1.505        |
|               | 16   | 0.06          | 0.261                 | 0.137     | 1.2      | 0.179        | 73        | 0.38          | 1.652                 | 0.642     | 0.8      | 1.546        |
|               | 17   | 0.06          | 0.261                 | 0.146     | 1.1      | 0.193        | 74        | 0.4           | 1.739                 | 0.650     | 0.8      | 1.588        |
|               | 18   | 0.07          | 0.304                 | 0.155     | 1.2      | 0.207        | 75        | 0.4           | 1.739                 | 0.659     | 0.8      | 1.631        |
|               | 19   | 0.07          | 0.304                 | 0.164     | 1.1      | 0.222        | 76        | 0.44          | 1.913                 | 0.668     | 0.7      | 1.676        |
|               | 20   | 0.07          | 0.304                 | 0.173     | 1.1      | 0.237        | 77        | 0.45          | 1.957                 | 0.677     | 0.7      | 1.721        |
|               | 21   | 0.08          | 0.348                 | 0.181     | 1.2      | 0.252        | 78        | 0.46          | 2.000                 | 0.686     | 0.7      | 1.768        |
|               | 22   | 0.08          | 0.348                 | 0.190     | 1.1      | 0.267        | 79        | 0.47          | 2.043                 | 0.695     | 0.8      | 1.817        |
|               | 23   | 0.08          | 0.348                 | 0.199     | 1.1      | 0.282        | 80        | 0.48          | 2.087                 | 0.704     | 0.8      | 1.867        |
|               | 24   | 0.08          | 0.348                 | 0.208     | 1.0      | 0.298        | 81        | 0.53          | 2.304                 | 0.712     | 0.7      | 1.919        |
|               | 25   | 0.08          | 0.348                 | 0.217     | 1.0      | 0.314        | 82        | 0.53          | 2.304                 | 0.721     | 0.7      | 1.972        |
|               | 26   | 0.09          | 0.391                 | 0.226     | 1.1      | 0.330        | 83        | 0.53          | 2.304                 | 0.730     | 0.8      | 2.028        |
|               | 27   | 0.1           | 0.435                 | 0.235     | 1.1      | 0.347        | 84        | 0.53          | 2.304                 | 0.739     | 0.8      | 2.085        |
|               | 28   | 0.1           | 0.435                 | 0.243     | 1.1      | 0.364        | 85        | 0.57          | 2.478                 | 0.748     | 0.8      | 2.145        |
|               | 29   | 0.1           | 0.435                 | 0.252     | 1.0      | 0.381        | 86        | 0.58          | 2.522                 | 0.757     | 0.8      | 2.207        |
|               | 30   | 0.1           | 0.435                 | 0.261     | 1.0      | 0.398        | 87        | 0.66          | 2.870                 | 0.765     | 0.7      | 2.271        |
|               | 31   | 0.11          | 0.478                 | 0.270     | 1.1      | 0.416        | 88        | 0.69          | 3.000                 | 0.774     | 0.7      | 2.338        |
|               | 32   | 0.11          | 0.478                 | 0.279     | 1.0      | 0.434        | 89        | 0.7           | 3.043                 | 0.783     | 0.7      | 2.408        |
|               | 33   | 0.11          | 0.478                 | 0.288     | 1.0      | 0.452        | 90        | 0.71          | 3.087                 | 0.792     | 0.7      | 2.481        |
|               | 34   | 0.12          | 0.522                 | 0.296     | 1.0      | 0.470        | 91        | 0.73          | 3.174                 | 0.801     | 0.7      | 2.558        |
|               | 35   | 0.13          | 0.565                 | 0.305     | 1.1      | 0.489        | 92        | 0.74          | 3.217                 | 0.810     | 0.7      | 2.638        |
|               | 36   | 0.13          | 0.565                 | 0.314     | 1.1      | 0.508        | 93        | 0.75          | 3.261                 | 0.819     | 0.8      | 2.722        |
|               | 37   | 0.13          | 0.565                 | 0.323     | 1.0      | 0.528        | 94        | 0.77          | 3.348                 | 0.827     | 0.8      | 2.811        |
|               | 38   | 0.13          | 0.565                 | 0.332     | 0.9      | 0.548        | 95        | 0.85          | 3.696                 | 0.836     | 0.7      | 2.904        |
|               | 39   | 0.13          | 0.565                 | 0.341     | 0.9      | 0.568        | 96        | 0.85          | 3.696                 | 0.845     | 0.8      | 3.004        |
|               | 40   | 0.14          | 0.609                 | 0.350     | 1.0      | 0.588        | 97        | 0.87          | 3.783                 | 0.854     | 0.8      | 3.109        |
|               | 41   | 0.14          | 0.609                 | 0.358     | 0.9      | 0.609        | 98        | 0.87          | 3.783                 | 0.863     | 0.8      | 3.222        |
|               | 42   | 0.15          | 0.652                 | 0.367     | 1.0      | 0.631        | 99        | 0.89          | 3.870                 | 0.872     | 0.8      | 3.342        |
|               | 43   | 0.15          | 0.652                 | 0.376     | 0.9      | 0.652        | 100       | 0.9           | 3.913                 | 0.881     | 0.8      | 3.472        |
|               | 44   | 0.15          | 0.652                 | 0.385     | 0.8      | 0.674        | 101       | 0.96          | 4.174                 | 0.889     | 0.8      | 3.612        |
|               | 45   | 0.16          | 0.696                 | 0.394     | 0.9      | 0.696        | 102       | 1.07          | 4.652                 | 0.898     | 0.8      | 3.764        |
|               | 46   | 0.17          | 0.739                 | 0.403     | 1.0      | 0.719        | 103       | 1.16          | 5.043                 | 0.907     | 0.8      | 3.931        |
|               | 47   | 0.17          | 0.739                 | 0.412     | 0.9      | 0.742        | 104       | 1.2           | 5.217                 | 0.916     | 0.8      | 4.115        |
|               | 48   | 0.18          | 0.783                 | 0.420     | 1.0      | 0.766        | 105       | 1.42          | 6.174                 | 0.925     | 0.7      | 4.321        |
|               | 49   | 0.18          | 0.783                 | 0.429     | 0.9      | 0.790        | 106       | 1.46          | 6.348                 | 0.934     | 0.7      | 4.554        |
|               | 50   | 0.19          | 0.826                 | 0.438     | 1.0      | 0.815        | 107       | 1.67          | 7.261                 | 0.942     | 0.7      | 4.822        |
|               | 51   | 0.21          | 0.913                 | 0.447     | 1.7      |              | 108       | 1.79          | 7.783                 | 0.951     | 0.7      | 5.136        |
|               | 52   | 0.22          | 0.957                 | 0.456     |          |              | 109       | 3.21          | 13.957                | 0.960     | 0.6      | 5.516        |
|               | 53   | 0.22          | 0.957                 | 0.465     |          |              | 110       | 3.38          | 14.696                | 0.969     | 0.6      | 5.996        |
|               | 54   | 0.22          | 0.957                 | 0.473     |          |              | 111       | 4.47          | 19.435                | 0.978     | 0.6      | 6.645        |
|               | 55   | 0.23          | 1.000                 | 0.482     |          |              | 112       | 5.38          | 23.391                | 0.987     | 0.6      | 7.641        |
|               | 56   | 0.23          | 1.000                 | 0.491     |          |              | 113       | 8.55          | 37.174                | 0.996     | 0.6      | 9.828        |
|               | 57   | 0.23          | 1.000                 | 0.500     |          |              |           |               |                       |           |          | 1.000        |



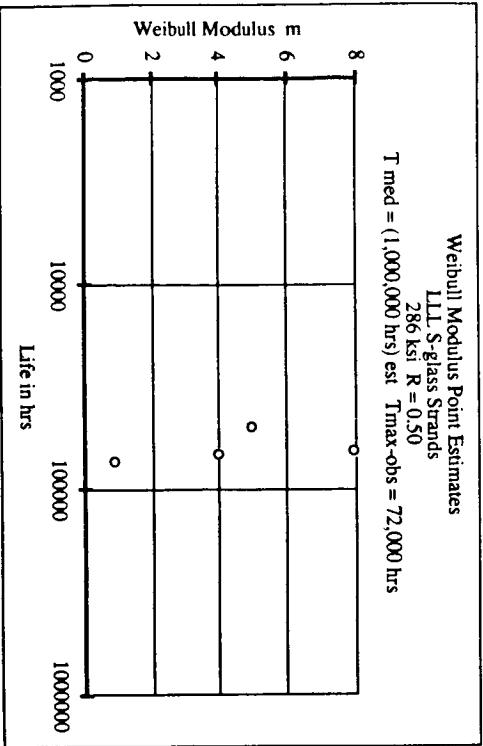
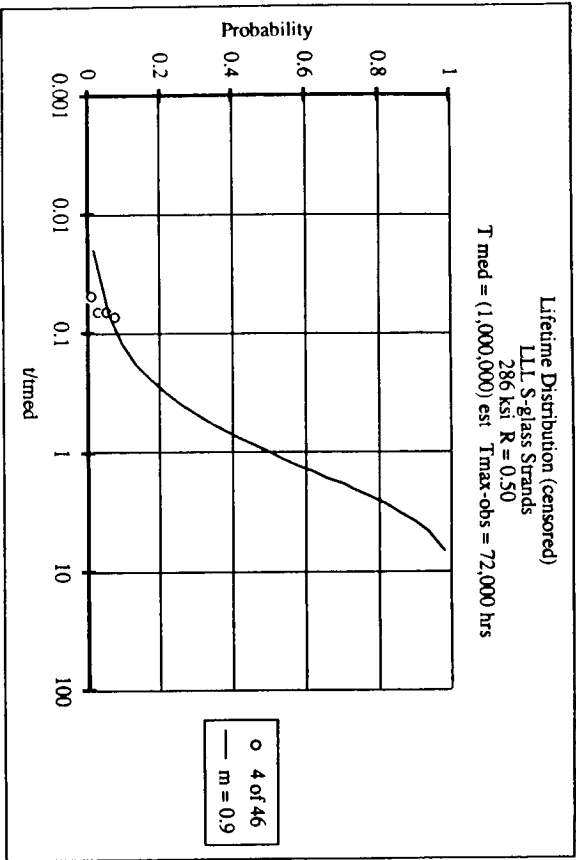
| LLL S-glass  |        |           |       |     |         |      |        |           |       |
|--------------|--------|-----------|-------|-----|---------|------|--------|-----------|-------|
| Rank         | Life t | Normal    | Prob  | est | fit     | Rank | Life t | Normal    | Prob  |
| r            | hrs    | $t_{med}$ | F     | b   | $b=0.9$ | r    | hrs    | $t_{med}$ | F     |
| Stress = 426 |        |           |       |     |         |      |        |           |       |
| 1            | 0.2    | 0.031     | 0.005 | 1.4 | 0.004   | 52   | 6.6    | 1.015     | 0.510 |
| 2            | 0.4    | 0.062     | 0.015 | 1.4 | 0.014   | 53   | 7.3    | 1.123     | 0.520 |
| 3            | 0.6    | 0.092     | 0.025 | 1.4 | 0.025   | 54   | 7.4    | 1.138     | 0.530 |
| 4            | 0.6    | 0.092     | 0.035 | 1.2 | 0.037   | 55   | 7.4    | 1.138     | 0.540 |
| 5            | 0.6    | 0.092     | 0.045 | 1.1 | 0.049   | 56   | 7.7    | 1.185     | 0.550 |
| 6            | 0.7    | 0.108     | 0.054 | 1.1 | 0.061   | 57   | 8      | 1.231     | 0.559 |
| 7            | 0.7    | 0.108     | 0.064 | 1.1 | 0.074   | 58   | 8.2    | 1.262     | 0.569 |
| 8            | 0.7    | 0.108     | 0.074 | 1.0 | 0.087   | 59   | 9.2    | 1.415     | 0.579 |
| 9            | 0.9    | 0.138     | 0.084 | 1.0 | 0.101   | 60   | 9.5    | 1.462     | 0.589 |
| 10           | 1      | 0.154     | 0.094 | 1.0 | 0.115   | 61   | 9.9    | 1.523     | 0.599 |
| 11           | 1      | 0.154     | 0.104 | 1.0 | 0.129   | 62   | 10.1   | 1.554     | 0.609 |
| 12           | 1.2    | 0.185     | 0.114 | 1.0 | 0.144   | 63   | 10.4   | 1.600     | 0.619 |
| 13           | 1.2    | 0.185     | 0.124 | 1.0 | 0.159   | 64   | 11.1   | 1.708     | 0.629 |
| 14           | 1.3    | 0.200     | 0.134 | 1.0 | 0.174   | 65   | 11.2   | 1.723     | 0.639 |
| 15           | 1.3    | 0.200     | 0.144 | 0.9 | 0.189   | 66   | 11.6   | 1.785     | 0.649 |
| 16           | 1.3    | 0.200     | 0.153 | 0.9 | 0.205   | 67   | 11.7   | 1.800     | 0.658 |
| 17           | 1.3    | 0.200     | 0.163 | 0.8 | 0.221   | 68   | 12.1   | 1.862     | 0.668 |
| 18           | 1.4    | 0.215     | 0.173 | 0.8 | 0.238   | 69   | 12.3   | 1.892     | 0.678 |
| 19           | 1.4    | 0.215     | 0.183 | 0.8 | 0.255   | 70   | 12.9   | 1.985     | 0.688 |
| 20           | 1.4    | 0.215     | 0.193 | 0.8 | 0.272   | 71   | 13.1   | 2.015     | 0.698 |
| 21           | 1.6    | 0.246     | 0.203 | 0.8 | 0.289   | 72   | 13.4   | 2.062     | 0.708 |
| 22           | 1.6    | 0.246     | 0.213 | 0.8 | 0.307   | 73   | 13.9   | 2.062     | 0.718 |
| 23           | 1.6    | 0.246     | 0.223 | 0.7 | 0.325   | 74   | 13.4   | 2.138     | 0.728 |
| 24           | 1.7    | 0.262     | 0.233 | 0.7 | 0.343   | 75   | 14.8   | 2.277     | 0.738 |
| 25           | 1.8    | 0.277     | 0.243 | 0.7 | 0.362   | 76   | 15.1   | 2.323     | 0.748 |
| 26           | 1.8    | 0.277     | 0.252 | 0.7 | 0.381   | 77   | 15.1   | 2.323     | 0.757 |
| 27           | 2.3    | 0.354     | 0.262 | 0.8 | 0.401   | 78   | 19.5   | 3.000     | 0.767 |
| 28           | 2.3    | 0.354     | 0.272 | 0.8 | 0.420   | 79   | 20.2   | 3.108     | 0.777 |
| 29           | 2.8    | 0.431     | 0.282 | 0.9 | 0.441   | 80   | 20.7   | 3.185     | 0.787 |
| 30           | 2.9    | 0.446     | 0.292 | 0.9 | 0.461   | 81   | 20.9   | 3.215     | 0.797 |
| 31           | 3      | 0.462     | 0.302 | 0.8 | 0.482   | 82   | 21.7   | 3.338     | 0.807 |
| 32           | 3      | 0.462     | 0.312 | 0.8 | 0.504   | 83   | 22.2   | 3.415     | 0.817 |
| 33           | 3.1    | 0.477     | 0.322 | 0.8 | 0.525   | 84   | 22.8   | 3.508     | 0.827 |
| 34           | 3.2    | 0.492     | 0.332 | 0.8 | 0.547   | 85   | 23.5   | 3.615     | 0.837 |
| 35           | 3.4    | 0.523     | 0.342 | 0.8 | 0.570   | 86   | 23.7   | 3.646     | 0.847 |
| 36           | 3.7    | 0.569     | 0.351 | 0.8 | 0.593   | 87   | 24.6   | 3.785     | 0.856 |
| 37           | 4.3    | 0.662     | 0.361 | 1.1 | 0.616   | 88   | 28.8   | 4.431     | 0.866 |
| 38           | 4.8    | 0.738     | 0.371 | 1.3 | 0.640   | 89   | 29.8   | 4.585     | 0.876 |
| 39           | 5      | 0.769     | 0.381 | 1.4 | 0.665   | 90   | 34     | 5.231     | 0.886 |
| 40           | 5.1    | 0.785     | 0.391 | 1.4 | 0.690   | 91   | 34.6   | 5.323     | 0.896 |
| 41           | 5.2    | 0.800     | 0.401 | 1.4 | 0.715   | 92   | 40     | 6.154     | 0.906 |
| 42           | 5.2    | 0.800     | 0.411 | 1.2 | 0.741   | 93   | 42.1   | 6.477     | 0.916 |
| 43           | 5.4    | 0.831     | 0.421 | 1.3 | 0.767   | 94   | 50.3   | 7.738     | 0.926 |
| 44           | 5.5    | 0.846     | 0.431 | 1.2 | 0.794   | 95   | 53.5   | 8.231     | 0.936 |
| 45           | 5.8    | 0.892     | 0.441 | 1.2 | 0.822   | 96   | 55     | 8.462     | 0.946 |
| 46           | 6      | 0.923     | 0.450 | 1.2 | 0.850   | 97   | 72     | 11.077    | 0.955 |
| 47           | 6.1    | 0.938     | 0.460 | 1.2 | 0.879   | 98   | 92.3   | 14.200    | 0.965 |
| 48           | 6.2    | 0.954     | 0.470 | 1.2 | 0.908   | 99   | 92.7   | 14.262    | 0.975 |
| 49           | 6.4    | 0.985     | 0.480 | 1.2 | 0.938   | 100  | 99.3   | 15.277    | 0.985 |
| 50           | 6.5    | 1.000     | 0.490 | 1.2 | 0.969   | 101  | 115.7  | 17.800    | 0.995 |
| 51           | 6.5    | 1.000     | 0.500 | 1.2 | 1.000   |      |        |           |       |



| LLL S-glass  |  | Rank Life t |     | Normal           |       | Prob |       | est |                  | fit   |       | Rank Life t |       | Normal           |      | Prob  |       | est |                  | fit |      |       |       |     |       |    |      |       |       |     |       |
|--------------|--|-------------|-----|------------------|-------|------|-------|-----|------------------|-------|-------|-------------|-------|------------------|------|-------|-------|-----|------------------|-----|------|-------|-------|-----|-------|----|------|-------|-------|-----|-------|
| Stress = 373 |  | r           | hrs | t <sub>med</sub> | F     | b    | b=0.9 | r   | t <sub>med</sub> | F     | b     | b=0.9       | r     | t <sub>med</sub> | F    | b     | b=0.9 | r   | t <sub>med</sub> | F   | b    | b=0.9 |       |     |       |    |      |       |       |     |       |
| R = 0.65     |  | 1           | 8   | 0.016            | 0.005 | 1.2  | 0.004 | 54  | 533              | 1.039 | 0.510 | 0.7         | 1.031 | 55               | 537  | 1.047 | 0.519 | 1.2 | 1.062            | 56  | 563  | 1.097 | 0.529 | 0.9 | 1.095 | 57 | 570  | 1.111 | 0.538 | 1.0 | 1.128 |
| N = 105      |  | 2           | 20  | 0.039            | 0.014 | 1.2  | 0.013 | 55  | 557              | 1.047 | 0.519 | 1.2         | 1.062 | 56               | 563  | 1.097 | 0.529 | 0.9 | 1.095            | 57  | 570  | 1.111 | 0.538 | 1.0 | 1.128 | 58 | 596  | 1.162 | 0.548 | 0.9 | 1.162 |
| Median = 513 |  | 3           | 35  | 0.049            | 0.024 | 1.1  | 0.024 | 56  | 563              | 1.097 | 0.529 | 0.9         | 1.095 | 57               | 570  | 1.111 | 0.538 | 1.0 | 1.128            | 58  | 596  | 1.162 | 0.548 | 0.9 | 1.162 | 59 | 643  | 1.253 | 0.557 | 0.7 | 1.196 |
| Type 1 (572) |  | 4           | 42  | 0.076            | 0.043 | 1.1  | 0.046 | 58  | 596              | 1.162 | 0.548 | 0.9         | 1.162 | 59               | 643  | 1.253 | 0.557 | 0.7 | 1.196            | 60  | 655  | 1.277 | 0.567 | 0.8 | 1.232 | 61 | 690  | 1.345 | 0.576 | 0.7 | 1.268 |
|              |  | 5           | 39  | 0.076            | 0.043 | 1.1  | 0.046 | 59  | 643              | 1.253 | 0.557 | 0.7         | 1.268 | 60               | 655  | 1.277 | 0.567 | 0.8 | 1.232            | 61  | 690  | 1.345 | 0.576 | 0.7 | 1.306 | 62 | 707  | 1.378 | 0.586 | 0.7 | 1.344 |
|              |  | 6           | 42  | 0.082            | 0.052 | 1.0  | 0.058 | 60  | 655              | 1.277 | 0.567 | 0.8         | 1.232 | 61               | 690  | 1.345 | 0.576 | 0.7 | 1.306            | 62  | 707  | 1.378 | 0.586 | 0.7 | 1.344 | 63 | 729  | 1.421 | 0.595 | 0.8 | 1.383 |
|              |  | 7           | 45  | 0.088            | 0.062 | 1.0  | 0.071 | 60  | 655              | 1.277 | 0.567 | 0.8         | 1.232 | 61               | 690  | 1.345 | 0.576 | 0.7 | 1.306            | 62  | 707  | 1.378 | 0.586 | 0.7 | 1.344 | 63 | 729  | 1.421 | 0.595 | 0.8 | 1.383 |
|              |  | 8           | 69  | 0.135            | 0.071 | 1.1  | 0.096 | 62  | 707              | 1.378 | 0.586 | 0.7         | 1.344 | 63               | 729  | 1.421 | 0.595 | 0.8 | 1.383            | 64  | 730  | 1.423 | 0.605 | 0.8 | 1.424 | 65 | 770  | 1.501 | 0.614 | 0.8 | 1.465 |
|              |  | 9           | 87  | 0.170            | 0.081 | 1.2  | 0.110 | 63  | 729              | 1.421 | 0.595 | 0.8         | 1.424 | 64               | 730  | 1.423 | 0.605 | 0.8 | 1.465            | 65  | 770  | 1.501 | 0.614 | 0.8 | 1.465 | 66 | 823  | 1.604 | 0.624 | 0.7 | 1.465 |
|              |  | 10          | 98  | 0.191            | 0.090 | 1.2  | 0.110 | 63  | 729              | 1.421 | 0.595 | 0.8         | 1.424 | 64               | 730  | 1.423 | 0.605 | 0.8 | 1.465            | 65  | 770  | 1.501 | 0.614 | 0.8 | 1.465 | 66 | 823  | 1.604 | 0.624 | 0.7 | 1.465 |
|              |  | 11          | 103 | 0.201            | 0.100 | 1.2  | 0.123 | 64  | 730              | 1.423 | 0.605 | 0.8         | 1.465 | 65               | 770  | 1.501 | 0.614 | 0.8 | 1.465            | 66  | 823  | 1.604 | 0.624 | 0.7 | 1.465 | 67 | 830  | 1.618 | 0.633 | 0.8 | 1.508 |
|              |  | 12          | 110 | 0.214            | 0.110 | 1.2  | 0.137 | 65  | 730              | 1.423 | 0.605 | 0.8         | 1.465 | 66               | 823  | 1.604 | 0.624 | 0.7 | 1.465            | 67  | 830  | 1.618 | 0.633 | 0.8 | 1.508 | 68 | 842  | 1.641 | 0.643 | 0.8 | 1.552 |
|              |  | 13          | 115 | 0.224            | 0.119 | 1.1  | 0.151 | 66  | 823              | 1.604 | 0.624 | 0.7         | 1.465 | 67               | 830  | 1.618 | 0.633 | 0.8 | 1.508            | 68  | 842  | 1.641 | 0.643 | 0.8 | 1.552 | 69 | 855  | 1.667 | 0.652 | 0.8 | 1.598 |
|              |  | 14          | 123 | 0.240            | 0.129 | 1.1  | 0.166 | 67  | 830              | 1.618 | 0.633 | 0.8         | 1.508 | 68               | 842  | 1.641 | 0.643 | 0.8 | 1.552            | 69  | 855  | 1.667 | 0.652 | 0.8 | 1.598 | 70 | 866  | 1.688 | 0.671 | 0.9 | 1.692 |
|              |  | 15          | 123 | 0.240            | 0.129 | 1.1  | 0.166 | 68  | 842              | 1.641 | 0.643 | 0.8         | 1.552 | 69               | 855  | 1.667 | 0.652 | 0.8 | 1.598            | 70  | 866  | 1.688 | 0.671 | 0.9 | 1.692 | 71 | 866  | 1.688 | 0.671 | 0.9 | 1.692 |
|              |  | 16          | 127 | 0.248            | 0.148 | 1.1  | 0.196 | 69  | 855              | 1.667 | 0.652 | 0.8         | 1.598 | 70               | 866  | 1.688 | 0.671 | 0.9 | 1.692            | 71  | 866  | 1.688 | 0.671 | 0.9 | 1.692 | 72 | 891  | 1.737 | 0.681 | 0.9 | 1.742 |
|              |  | 17          | 135 | 0.263            | 0.157 | 1.0  | 0.227 | 71  | 866              | 1.688 | 0.671 | 0.9         | 1.692 | 72               | 891  | 1.737 | 0.681 | 0.9 | 1.742            | 73  | 900  | 1.754 | 0.690 | 0.9 | 1.794 | 74 | 945  | 1.842 | 0.700 | 0.9 | 1.847 |
|              |  | 18          | 136 | 0.265            | 0.167 | 1.0  | 0.227 | 71  | 866              | 1.688 | 0.671 | 0.9         | 1.692 | 72               | 891  | 1.737 | 0.681 | 0.9 | 1.742            | 73  | 900  | 1.754 | 0.690 | 0.9 | 1.794 | 74 | 945  | 1.842 | 0.700 | 0.9 | 1.847 |
|              |  | 19          | 144 | 0.281            | 0.176 | 1.0  | 0.243 | 72  | 891              | 1.737 | 0.681 | 0.9         | 1.742 | 73               | 900  | 1.754 | 0.690 | 0.9 | 1.794            | 74  | 945  | 1.842 | 0.700 | 0.9 | 1.847 | 75 | 948  | 1.848 | 0.710 | 0.9 | 1.902 |
|              |  | 20          | 149 | 0.290            | 0.186 | 1.0  | 0.259 | 73  | 900              | 1.754 | 0.690 | 0.9         | 1.794 | 74               | 945  | 1.842 | 0.700 | 0.9 | 1.847            | 75  | 948  | 1.848 | 0.710 | 0.9 | 1.902 | 76 | 1004 | 1.957 | 0.719 | 0.9 | 1.959 |
|              |  | 21          | 153 | 0.298            | 0.195 | 1.0  | 0.275 | 74  | 945              | 1.842 | 0.700 | 0.9         | 1.847 | 75               | 948  | 1.848 | 0.710 | 0.9 | 1.902            | 76  | 1004 | 1.957 | 0.719 | 0.9 | 1.959 | 77 | 1011 | 1.971 | 0.729 | 0.9 | 2.018 |
|              |  | 22          | 183 | 0.357            | 0.205 | 1.1  | 0.292 | 75  | 948              | 1.848 | 0.710 | 0.9         | 1.902 | 76               | 1004 | 1.957 | 0.719 | 0.9 | 1.959            | 77  | 1011 | 1.971 | 0.729 | 0.9 | 2.018 | 78 | 1029 | 2.006 | 0.738 | 0.9 | 2.080 |
|              |  | 23          | 190 | 0.370            | 0.214 | 1.1  | 0.309 | 76  | 1004             | 1.957 | 0.719 | 0.9         | 1.959 | 77               | 1011 | 1.971 | 0.729 | 0.9 | 2.018            | 78  | 1029 | 2.006 | 0.738 | 0.9 | 2.080 | 79 | 1063 | 2.072 | 0.748 | 0.9 | 2.144 |
|              |  | 24          | 196 | 0.382            | 0.224 | 1.0  | 0.327 | 77  | 1011             | 1.971 | 0.738 | 0.9         | 2.080 | 78               | 1029 | 2.006 | 0.738 | 0.9 | 2.080            | 79  | 1063 | 2.072 | 0.748 | 0.9 | 2.144 | 80 | 1205 | 2.349 | 0.757 | 0.8 | 2.210 |
|              |  | 25          | 202 | 0.394            | 0.233 | 1.0  | 0.345 | 78  | 1063             | 2.072 | 0.748 | 0.9         | 2.144 | 79               | 1063 | 2.072 | 0.748 | 0.9 | 2.144            | 80  | 1205 | 2.349 | 0.757 | 0.8 | 2.210 | 81 | 1226 | 2.390 | 0.767 | 0.9 | 2.280 |
|              |  | 26          | 206 | 0.402            | 0.243 | 1.0  | 0.363 | 79  | 1063             | 2.072 | 0.748 | 0.9         | 2.144 | 80               | 1205 | 2.349 | 0.757 | 0.8 | 2.210            | 81  | 1226 | 2.390 | 0.767 | 0.9 | 2.280 | 82 | 1297 | 2.528 | 0.776 | 0.8 | 2.353 |
|              |  | 27          | 209 | 0.407            | 0.252 | 1.0  | 0.381 | 80  | 1205             | 2.349 | 0.757 | 0.8         | 2.210 | 81               | 1226 | 2.390 | 0.767 | 0.9 | 2.280            | 82  | 1297 | 2.528 | 0.776 | 0.8 | 2.353 | 83 | 1308 | 2.550 | 0.786 | 0.9 | 2.429 |
|              |  | 28          | 218 | 0.425            | 0.262 | 1.0  | 0.400 | 81  | 1226             | 2.390 | 0.767 | 0.9         | 2.280 | 82               | 1297 | 2.528 | 0.776 | 0.8 | 2.353            | 83  | 1308 | 2.550 | 0.786 | 0.9 | 2.429 | 84 | 1334 | 2.600 | 0.795 | 0.9 | 2.508 |
|              |  | 29          | 222 | 0.433            | 0.271 | 0.9  | 0.419 | 82  | 1297             | 2.528 | 0.776 | 0.8         | 2.353 | 83               | 1308 | 2.550 | 0.786 | 0.9 | 2.508            | 84  | 1334 | 2.600 | 0.795 | 0.9 | 2.508 | 85 | 1379 | 2.688 | 0.805 | 0.9 | 2.592 |
|              |  | 30          | 231 | 0.450            | 0.281 | 0.9  | 0.438 | 83  | 1308             | 2.550 | 0.786 | 0.9         | 2.429 | 84               | 1334 | 2.600 | 0.795 | 0.9 | 2.508            | 85  | 1379 | 2.688 | 0.805 | 0.9 | 2.592 | 86 | 1381 | 2.692 | 0.814 | 0.9 | 2.681 |
|              |  | 31          | 237 | 0.460            | 0.290 | 0.9  | 0.458 | 84  | 1334             | 2.600 | 0.795 | 0.9         | 2.508 | 85               | 1379 | 2.688 | 0.805 | 0.9 | 2.592            | 86  | 1381 | 2.692 | 0.814 | 0.9 | 2.681 | 87 | 1403 | 2.735 | 0.843 | 1.0 | 2.978 |
|              |  | 32          | 238 | 0.464            | 0.300 | 0.9  | 0.478 | 85  | 1379             | 2.688 | 0.814 | 0.9         | 2.681 | 86               | 1403 | 2.735 | 0.843 | 1.0 | 2.978            | 87  | 1434 | 2.795 | 0.852 | 1.0 | 3.090 | 88 | 1454 | 2.841 | 0.862 | 1.0 | 3.210 |
|              |  | 33          | 240 | 0.468            | 0.310 | 0.8  | 0.498 | 86  | 1379             | 2.688 | 0.814 | 0.9         | 2.681 | 87               | 1434 | 2.795 | 0.852 | 1.0 | 3.090            | 88  | 1454 | 2.841 | 0.862 | 1.0 | 3.210 | 89 | 1460 | 2.860 | 0.871 | 1.0 | 3.339 |
|              |  | 34          | 274 | 0.534            | 0.319 | 0.9  | 0.519 | 87  | 1381             | 2.692 | 0.824 | 0.9         | 2.774 | 88               | 1391 | 2.712 | 0.833 | 1.0 | 2.873            | 89  | 1460 | 2.860 | 0.871 | 1.0 | 3.339 | 90 | 1478 | 2.906 | 0.881 | 1.0 | 3.478 |
|              |  | 35          | 305 | 0.595            | 0.329 | 1.1  | 0.540 | 88  | 1391             | 2.712 | 0.833 | 1.0         | 2.873 | 89               | 1460 | 2.860 | 0.871 | 1.0 | 3.339            | 90  | 1478 | 2.906 | 0.881 | 1.0 | 3.478 | 91 | 1600 | 3.119 | 0.871 | 1.0 | 3.339 |
|              |  | 36          | 318 | 0.620            | 0.338 | 1.1  | 0.562 | 89  | 1434             | 2.795 | 0.852 | 1.0         | 3.090 | 90               | 1454 | 2.841 | 0.862 | 1.0 | 3.210            | 91  | 1600 | 3.119 | 0.871 | 1.0 | 3.339 | 92 | 1630 | 3.177 | 0.881 | 1.0 | 3.478 |
|              |  | 37          | 335 | 0.653            | 0.348 | 1.1  | 0.584 | 90  | 1434             | 2.795 | 0.852 | 1.0         | 3.090 | 91               | 1600 | 3.119 | 0.871 | 1.0 | 3.339            | 92  | 1630 | 3.177 | 0.881 | 1.0 | 3.478 | 93 | 1630 | 3.177 | 0.881 | 1.0 | 3.478 |
|              |  | 38          | 341 | 0.665            | 0.357 | 1.1  | 0.606 | 91  | 1467             | 2.860 | 0.862 | 1.0         | 3.339 | 92               | 1600 | 3.119 | 0.871 | 1.0 | 3.339            | 93  | 1630 | 3.177 | 0.881 | 1.0 | 3.478 | 94 | 1648 | 3.212 | 0.890 | 1.0 | 3.630 |
|              |  | 39          | 368 | 0.717            | 0.367 | 1.3  | 0.629 | 92  | 1600             | 3.119 | 0.871 | 1.0         | 3.339 | 93               | 1630 | 3.177 | 0.881 | 1.0 | 3.478            | 94  | 1648 | 3.212 | 0.890 | 1.0 | 3.630 | 95 | 1786 | 3.481 | 0.900 | 1.0 | 3.796 |
|              |  | 40          | 374 | 0.729            | 0.376 | 1.2  | 0.652 | 93  | 1630             | 3.177 | 0.881 | 1.0         | 3.478 | 94               | 1648 | 3.212 | 0.890 | 1.0 | 3.630            | 95  | 1786 | 3.481 | 0.900 | 1.0 | 3.796 | 96 | 1894 | 3.692 | 0.910 | 1.0 | 3.980 |
|              |  | 41          | 391 | 0.762            | 0.386 | 1.3  | 0.676 | 94  | 1648             | 3.212 | 0.890 | 1.0         | 3.630 | 95               | 1786 | 3.481 | 0.900 | 1.0 | 3.796            | 96  | 1894 | 3.692 | 0.910 | 1.0 | 3.980 | 97 | 2152 | 4.195 | 0.919 | 0.9 | 4.185 |
|              |  | 42          | 413 | 0.805            | 0.395 | 1.5  | 0.700 | 95  | 1786             | 3.481 | 0.900 | 1.0         | 3.796 | 96               | 1894 | 3.692 | 0.910 | 1.0 | 3.980            | 97  | 2152 | 4.195 | 0.919 | 0.9 | 4.185 | 98 | 2262 | 4.409 | 0.929 | 0.9 | 4.417 |
|              |  | 43          | 419 | 0.817            | 0.405 | 1.4  | 0.725 | 96  | 1894             | 3.692 | 0.910 | 1.0         | 3.980 | 97               | 2152 | 4.195 | 0.919 | 0.9 | 4.185            | 98  | 2262 | 4.409 | 0.929 | 0.9 | 4.417 | 99 | 2395 | 4.669 | 0.938 | 0.9 | 4.684 |
|              |  | 44          | 426 | 0.830            | 0.414 | 1.4  | 0.755 | 97  | 2152             | 4.195 | 0.919 | 0.9         | 4.185 | 98               | 2262 | 4.409 |       |     |                  |     |      |       |       |     |       |    |      |       |       |     |       |



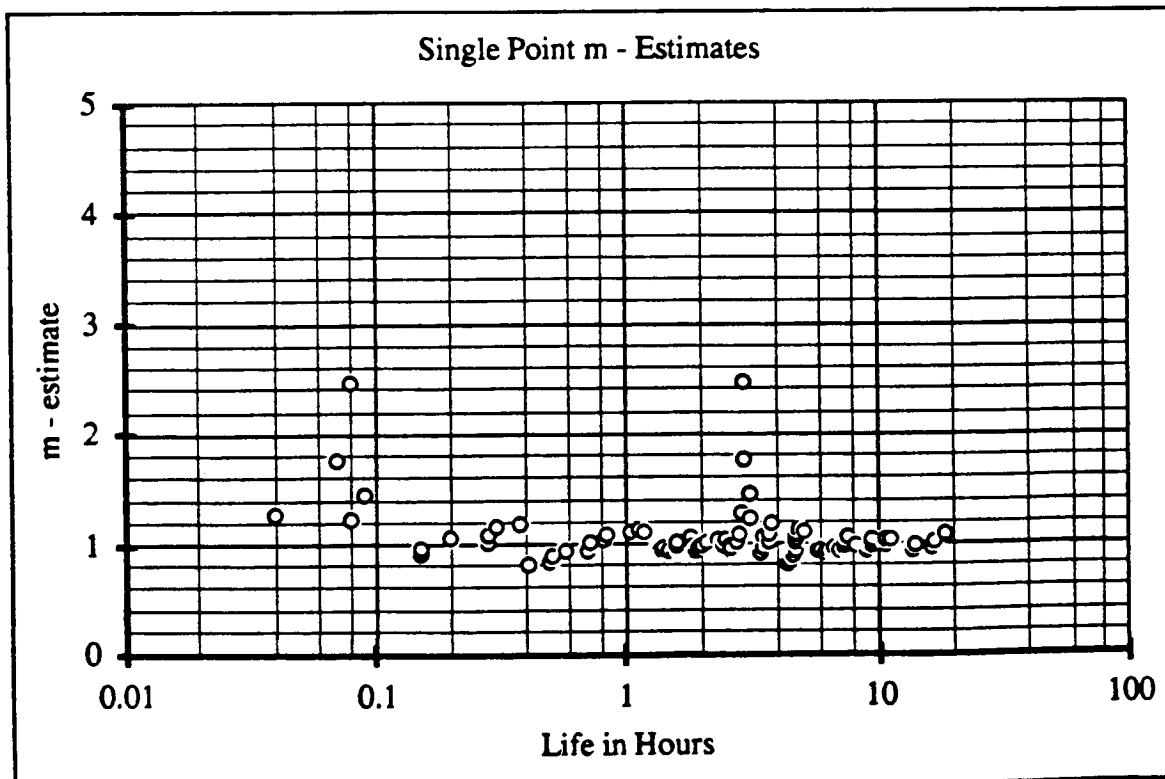
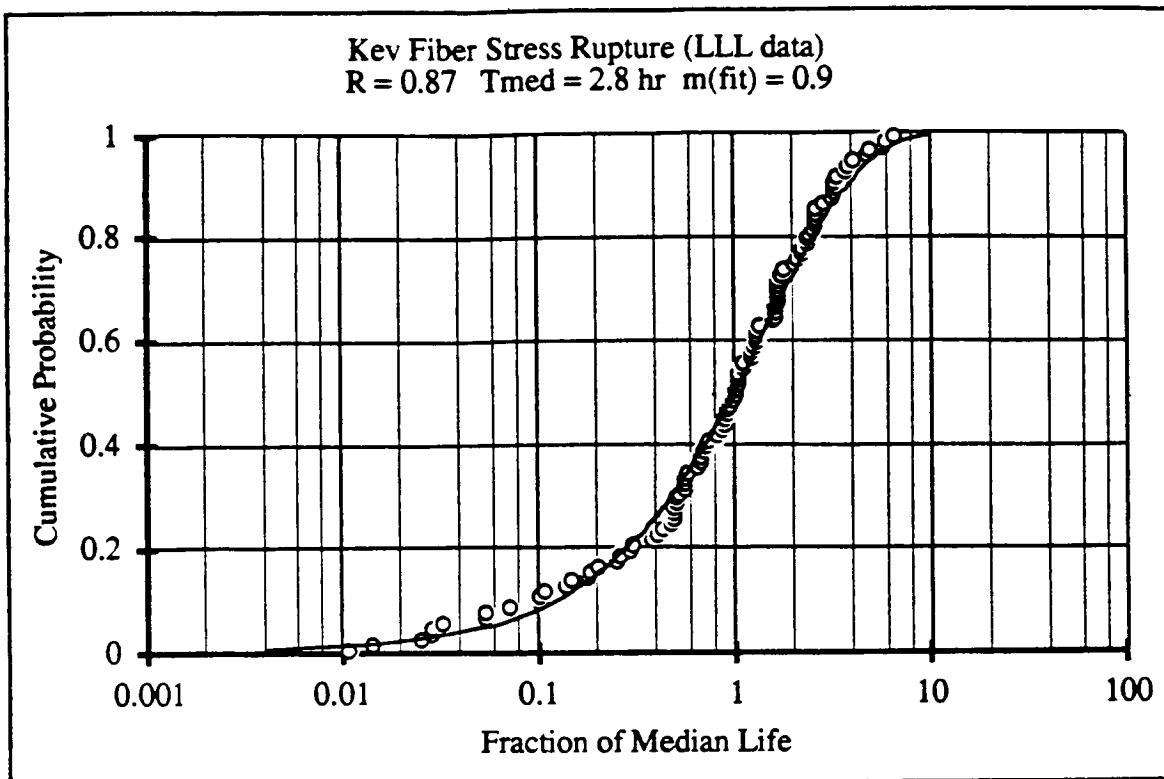
| LLL S-glass  |         |      |       |            |        |        |       |        |        |
|--------------|---------|------|-------|------------|--------|--------|-------|--------|--------|
| Stress =     | 332 ksi | Rank | 1     | Life t     | hrs    | Normal | Prob  | est    | fit    |
| R =          | 58%     | 2    | 745   | $V_{norm}$ | 0.010  | 0.0051 | 1.0   | 0.004  | sample |
| N =          | 98      | 3    | 914   | 0.012      | 0.0153 | 0.8    | 0.015 | 100811 | 100000 |
| Normal t =   | 76032   | 4    | 2712  | 0.036      | 0.0255 | 0.9    | 0.026 | 103938 | 0.009  |
| $S_{norm}$ = | 0.5929  | 5    | 3384  | 0.045      | 0.0357 | 0.9    | 0.038 | 103955 | 0.027  |
| Remaining:   |         | 6    | 4211  | 0.055      | 0.0459 | 0.8    | 0.050 | 105702 | 0.034  |
| N1 =         | 50      | 7    | 5376  | 0.071      | 0.0561 | 0.8    | 0.063 | 106753 | 0.054  |
| $t_i$ =      | 7200 hr | 8    | 5400  | 0.074      | 0.0663 | 0.8    | 0.077 | 109807 | 0.054  |
| AF =         | 0.02    | 9    | 5640  | 0.074      | 0.0765 | 0.7    | 0.090 | 112304 | 0.056  |
|              |         | 10   | 6816  | 0.090      | 0.0867 | 0.7    | 0.119 | 112145 | 0.068  |
|              |         | 11   | 7080  | 0.092      | 0.0969 | 0.7    | 0.114 | 114818 | 0.070  |
|              |         | 12   | 9000  | 0.093      | 0.1071 | 0.6    | 0.134 | 117839 | 0.071  |
|              |         | 13   | 9096  | 0.118      | 0.1271 | 0.6    | 0.164 | 118907 | 0.090  |
|              |         | 14   | 9840  | 0.120      | 0.1471 | 0.6    | 0.195 | 125840 | 0.091  |
|              |         | 15   | 14928 | 0.129      | 0.1671 | 0.5    | 0.228 | 131705 | 0.098  |
|              |         | 16   | 14928 | 0.196      | 0.1871 | 0.6    | 0.261 | 124906 | 0.149  |
|              |         | 17   | 25968 | 0.342      | 0.2071 | 0.8    | 0.297 | 110449 | 0.260  |
|              |         | 18   | 27504 | 0.362      | 0.2271 | 0.7    | 0.333 | 114063 | 0.275  |
|              |         | 19   | 27792 | 0.366      | 0.2471 | 1.4    | 0.410 | 121063 | 0.278  |
|              |         | 20   | 71328 | 0.686      | 0.2671 |        | 0.451 | 93272  | 0.522  |
|              |         | 21   | 76032 | 0.938      | 0.2871 |        | 0.493 |        | 0.713  |
|              |         |      |       | 1.000      | 0.3071 |        |       |        | 0.760  |



| LLL S-glass               |       | Rank | Life t | Normal            | Prob  | est | est      | est      | med      | Life est | med | Norm |
|---------------------------|-------|------|--------|-------------------|-------|-----|----------|----------|----------|----------|-----|------|
|                           |       | T    | hrs    | U <sub>norm</sub> | F     | m   | med life | life 0.9 | at 1e-06 | by 1E06  |     |      |
| Stress =                  | 286   | 1    | 48792  | 0.672             | 0.011 | 5.0 | 112443   | 4907270  | 7514     | 0.049    |     |      |
| R =                       | 0.5   | 2    | 65112  | 0.896             | 0.033 | 7.9 | 95516    | 1908427  | 17534    | 0.065    |     |      |
|                           |       | 3    | 66600  | 0.917             | 0.054 | 4.0 | 125193   | 1092816  | 4301     | 0.067    |     |      |
| N =                       | 46    | 4    | 72648  | 1.000             | 0.076 | 0.9 |          | 809811   | 0.26     | 0.073    |     |      |
| Normal t =                | 72648 |      |        |                   |       |     |          |          |          |          |     |      |
| S <sub>norm</sub> =       | 0.924 |      |        |                   |       |     |          |          |          |          |     |      |
| Type 1                    | (572) |      |        |                   |       |     |          |          |          |          |     |      |
| (Use) 1000000 and m = 0.9 |       |      |        |                   |       |     |          |          |          |          |     |      |

## **APPENDIX B**

### **Kevlar Composite Strand Data Tables and Plots Based on Reference 2**

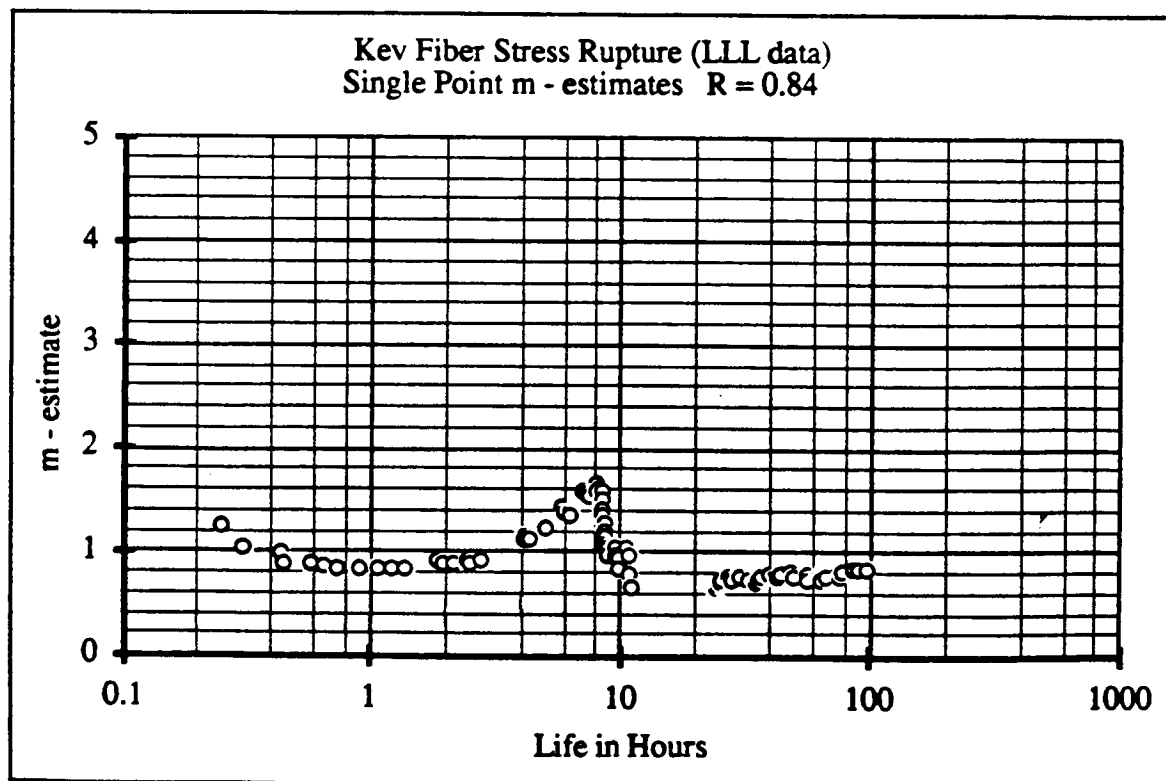
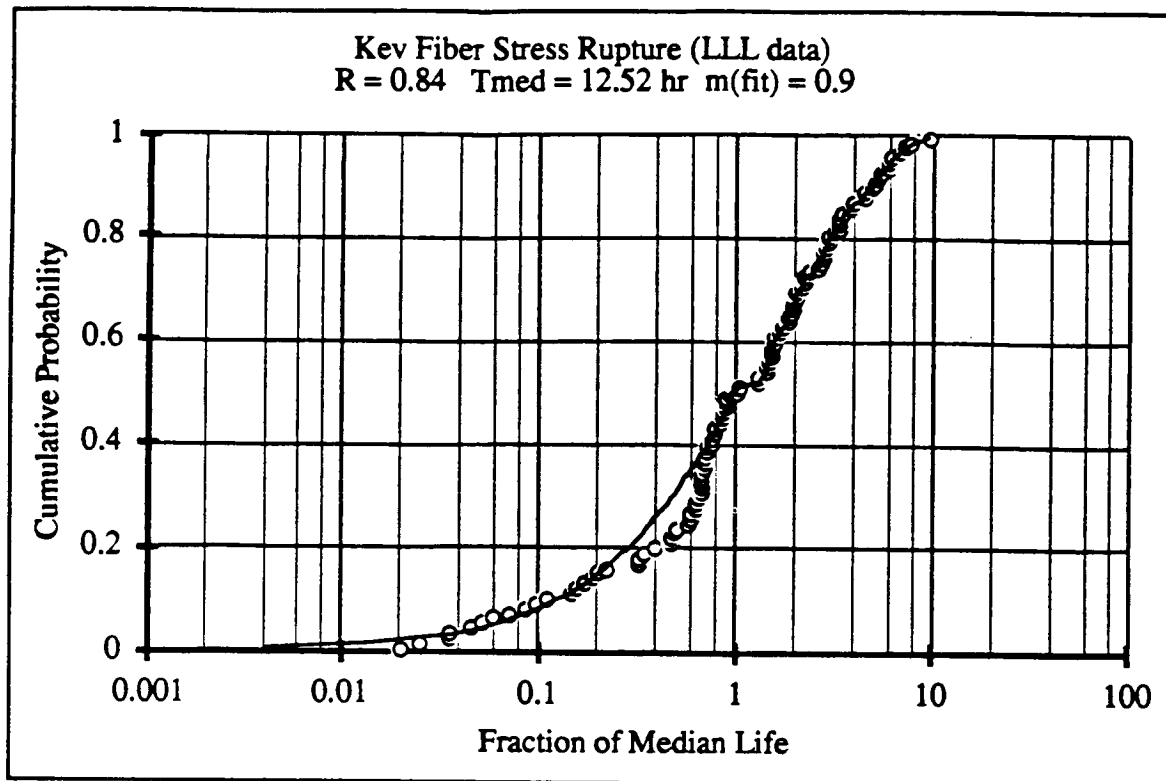


| UCID 19849<br>440 ksi 0.87<br>N = 100<br>Med = 2.795 |            |       |               |       |      |            |     |            |       |               |       |      |            |
|------------------------------------------------------|------------|-------|---------------|-------|------|------------|-----|------------|-------|---------------|-------|------|------------|
| r                                                    | Life - hrs | t/med | F = (r-1/2)/N | 1-F   | best | pred t/med | r   | Life - hrs | t/med | F = (r-1/2)/N | 1-F   | best | pred t/med |
| 1                                                    | 0.03       | 0.011 | 0.005         | 0.995 | 1.09 | 0.004      | 51  | 2.80       | 1.002 | 0.505         | 0.495 | 1.29 | 1.016      |
| 2                                                    | 0.04       | 0.014 | 0.015         | 0.985 | 0.90 | 0.014      | 52  | 2.89       | 1.034 | 0.515         | 0.485 | 1.77 | 1.049      |
| 3                                                    | 0.07       | 0.025 | 0.025         | 0.975 | 0.90 | 0.025      | 53  | 2.91       | 1.041 | 0.525         | 0.475 | 2.47 | 1.083      |
| 4                                                    | 0.08       | 0.029 | 0.035         | 0.965 | 0.84 | 0.037      | 54  | 2.91       | 1.041 | 0.535         | 0.465 | 2.47 | 1.117      |
| 5                                                    | 0.08       | 0.029 | 0.045         | 0.955 | 0.76 | 0.049      | 55  | 3.10       | 1.109 | 0.545         | 0.455 | 1.23 | 1.152      |
| 6                                                    | 0.09       | 0.032 | 0.055         | 0.945 | 0.73 | 0.062      | 56  | 3.11       | 1.113 | 0.555         | 0.445 | 1.46 | 1.188      |
| 7                                                    | 0.15       | 0.054 | 0.065         | 0.935 | 0.80 | 0.075      | 57  | 3.42       | 1.224 | 0.565         | 0.435 | 0.91 | 1.226      |
| 8                                                    | 0.15       | 0.054 | 0.075         | 0.925 | 0.75 | 0.088      | 58  | 3.47       | 1.242 | 0.575         | 0.425 | 0.97 | 1.264      |
| 9                                                    | 0.20       | 0.072 | 0.085         | 0.915 | 0.78 | 0.102      | 59  | 3.50       | 1.252 | 0.585         | 0.415 | 1.06 | 1.303      |
| 10                                                   | 0.28       | 0.100 | 0.095         | 0.905 | 0.84 | 0.116      | 60  | 3.64       | 1.302 | 0.595         | 0.405 | 1.00 | 1.343      |
| 11                                                   | 0.28       | 0.100 | 0.105         | 0.895 | 0.80 | 0.131      | 61  | 3.67       | 1.313 | 0.605         | 0.395 | 1.07 | 1.384      |
| 12                                                   | 0.30       | 0.107 | 0.115         | 0.885 | 0.78 | 0.145      | 62  | 3.68       | 1.317 | 0.615         | 0.385 | 1.16 | 1.427      |
| 13                                                   | 0.38       | 0.136 | 0.125         | 0.875 | 0.83 | 0.160      | 63  | 3.75       | 1.342 | 0.625         | 0.375 | 1.18 | 1.471      |
| 14                                                   | 0.41       | 0.147 | 0.135         | 0.865 | 0.81 | 0.176      | 64  | 4.39       | 1.571 | 0.635         | 0.365 | 0.83 | 1.516      |
| 15                                                   | 0.50       | 0.179 | 0.145         | 0.855 | 0.86 | 0.192      | 65  | 4.50       | 1.610 | 0.645         | 0.355 | 0.84 | 1.562      |
| 16                                                   | 0.51       | 0.182 | 0.155         | 0.845 | 0.83 | 0.208      | 66  | 4.53       | 1.621 | 0.655         | 0.345 | 0.89 | 1.610      |
| 17                                                   | 0.57       | 0.204 | 0.165         | 0.835 | 0.85 | 0.224      | 67  | 4.53       | 1.621 | 0.665         | 0.335 | 0.94 | 1.660      |
| 18                                                   | 0.70       | 0.250 | 0.175         | 0.825 | 0.93 | 0.241      | 68  | 4.65       | 1.664 | 0.675         | 0.325 | 0.95 | 1.711      |
| 19                                                   | 0.72       | 0.258 | 0.185         | 0.815 | 0.90 | 0.258      | 69  | 4.65       | 1.664 | 0.685         | 0.315 | 1.00 | 1.764      |
| 20                                                   | 0.82       | 0.293 | 0.195         | 0.805 | 0.95 | 0.275      | 70  | 4.69       | 1.678 | 0.695         | 0.305 | 1.04 | 1.819      |
| 21                                                   | 0.84       | 0.301 | 0.205         | 0.795 | 0.92 | 0.293      | 71  | 4.70       | 1.682 | 0.705         | 0.295 | 1.09 | 1.876      |
| 22                                                   | 1.05       | 0.376 | 0.215         | 0.785 | 1.07 | 0.311      | 72  | 4.75       | 1.699 | 0.715         | 0.285 | 1.12 | 1.934      |
| 23                                                   | 1.13       | 0.404 | 0.225         | 0.775 | 1.10 | 0.329      | 73  | 4.84       | 1.732 | 0.725         | 0.275 | 1.13 | 1.996      |
| 24                                                   | 1.19       | 0.426 | 0.235         | 0.765 | 1.11 | 0.348      | 74  | 5.01       | 1.792 | 0.735         | 0.265 | 1.11 | 2.059      |
| 25                                                   | 1.32       | 0.472 | 0.245         | 0.755 | 1.20 | 0.367      | 75  | 5.71       | 2.043 | 0.745         | 0.255 | 0.95 | 2.126      |
| 26                                                   | 1.35       | 0.483 | 0.255         | 0.745 | 1.18 | 0.386      | 76  | 5.92       | 2.118 | 0.755         | 0.245 | 0.94 | 2.195      |
| 27                                                   | 1.35       | 0.483 | 0.265         | 0.735 | 1.12 | 0.406      | 77  | 6.19       | 2.215 | 0.765         | 0.235 | 0.93 | 2.268      |
| 28                                                   | 1.36       | 0.487 | 0.275         | 0.725 | 1.07 | 0.426      | 78  | 6.33       | 2.265 | 0.775         | 0.225 | 0.94 | 2.343      |
| 29                                                   | 1.39       | 0.497 | 0.285         | 0.715 | 1.04 | 0.446      | 79  | 6.59       | 2.358 | 0.785         | 0.215 | 0.93 | 2.423      |
| 30                                                   | 1.40       | 0.501 | 0.295         | 0.705 | 0.99 | 0.467      | 80  | 6.62       | 2.369 | 0.795         | 0.205 | 0.96 | 2.506      |
| 31                                                   | 1.48       | 0.530 | 0.305         | 0.695 | 1.01 | 0.489      | 81  | 6.94       | 2.483 | 0.805         | 0.195 | 0.94 | 2.594      |
| 32                                                   | 1.55       | 0.555 | 0.315         | 0.685 | 1.03 | 0.510      | 82  | 7.16       | 2.562 | 0.815         | 0.185 | 0.95 | 2.687      |
| 33                                                   | 1.56       | 0.558 | 0.325         | 0.675 | 0.97 | 0.532      | 83  | 7.29       | 2.608 | 0.825         | 0.175 | 0.96 | 2.786      |
| 34                                                   | 1.57       | 0.562 | 0.335         | 0.665 | 0.92 | 0.555      | 84  | 7.36       | 2.633 | 0.835         | 0.165 | 0.99 | 2.891      |
| 35                                                   | 1.59       | 0.569 | 0.345         | 0.655 | 0.88 | 0.578      | 85  | 7.38       | 2.640 | 0.845         | 0.155 | 1.02 | 3.002      |
| 36                                                   | 1.80       | 0.644 | 0.355         | 0.645 | 1.04 | 0.601      | 86  | 7.42       | 2.655 | 0.855         | 0.145 | 1.05 | 3.122      |
| 37                                                   | 1.85       | 0.662 | 0.365         | 0.635 | 1.02 | 0.625      | 87  | 8.09       | 2.894 | 0.865         | 0.135 | 1.00 | 3.250      |
| 38                                                   | 1.86       | 0.665 | 0.375         | 0.625 | 0.95 | 0.649      | 88  | 8.88       | 3.177 | 0.875         | 0.125 | 0.95 | 3.389      |
| 39                                                   | 1.92       | 0.687 | 0.385         | 0.615 | 0.94 | 0.674      | 89  | 9.05       | 3.238 | 0.885         | 0.115 | 0.97 | 3.541      |
| 40                                                   | 2.00       | 0.716 | 0.395         | 0.605 | 0.96 | 0.700      | 90  | 9.25       | 3.309 | 0.895         | 0.105 | 0.99 | 3.707      |
| 41                                                   | 2.08       | 0.744 | 0.405         | 0.595 | 0.98 | 0.725      | 91  | 9.26       | 3.313 | 0.905         | 0.095 | 1.02 | 3.890      |
| 42                                                   | 2.27       | 0.812 | 0.415         | 0.585 | 1.23 | 0.752      | 92  | 9.47       | 3.388 | 0.915         | 0.085 | 1.04 | 4.095      |
| 43                                                   | 2.38       | 0.852 | 0.425         | 0.575 | 1.40 | 0.779      | 93  | 10.50      | 3.757 | 0.925         | 0.075 | 1.00 | 4.326      |
| 44                                                   | 2.46       | 0.880 | 0.435         | 0.565 | 1.52 | 0.806      | 94  | 10.57      | 3.782 | 0.935         | 0.065 | 1.03 | 4.593      |
| 45                                                   | 2.49       | 0.891 | 0.445         | 0.555 | 1.41 | 0.834      | 95  | 11.25      | 4.025 | 0.945         | 0.055 | 1.03 | 4.906      |
| 46                                                   | 2.61       | 0.934 | 0.455         | 0.545 | 1.94 | 0.863      | 96  | 13.49      | 4.826 | 0.955         | 0.045 | 0.95 | 5.284      |
| 47                                                   | 2.61       | 0.934 | 0.465         | 0.535 | 1.50 | 0.892      | 97  | 13.78      | 4.930 | 0.965         | 0.035 | 0.99 | 5.762      |
| 48                                                   | 2.62       | 0.937 | 0.475         | 0.525 | 1.13 | 0.922      | 98  | 16.15      | 5.778 | 0.975         | 0.025 | 0.95 | 6.408      |
| 49                                                   | 2.74       | 0.980 | 0.485         | 0.515 | 2.19 | 0.953      | 99  | 16.59      | 5.936 | 0.985         | 0.015 | 1.01 | 7.402      |
| 50                                                   | 2.79       | 0.998 | 0.495         | 0.505 |      | 0.984      | 100 | 18.16      | 6.497 | 0.995         | 0.005 | 1.09 | 9.582      |

more

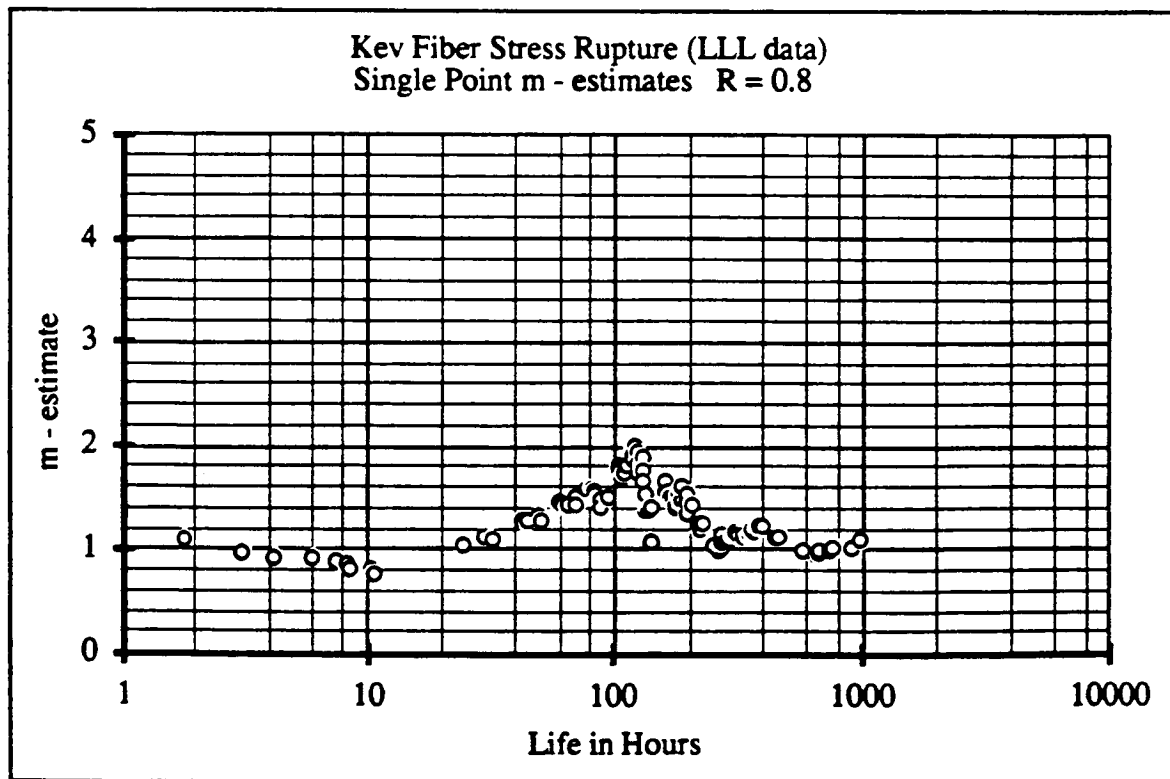
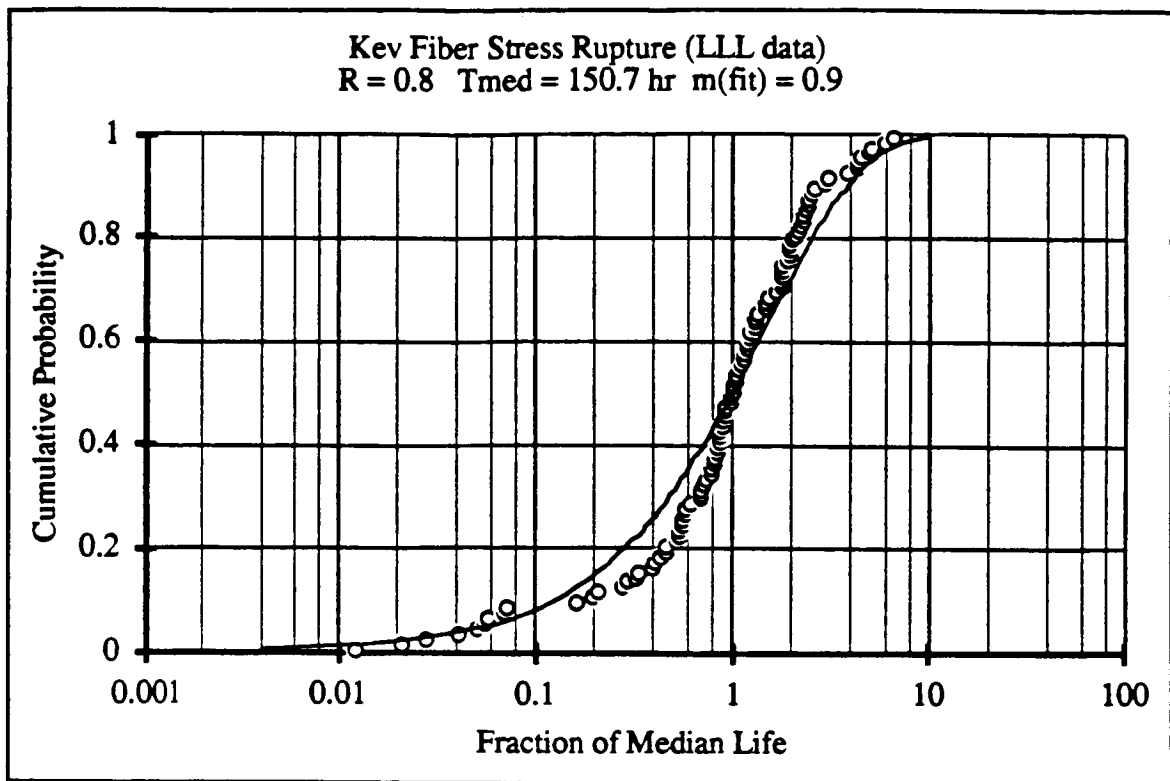
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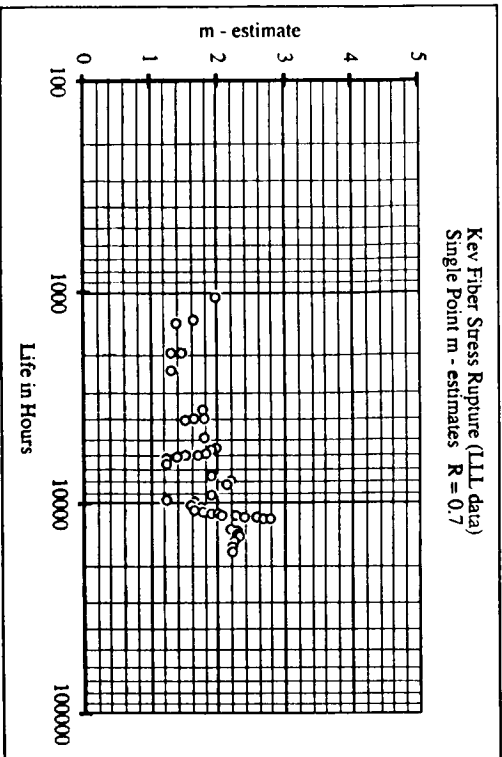
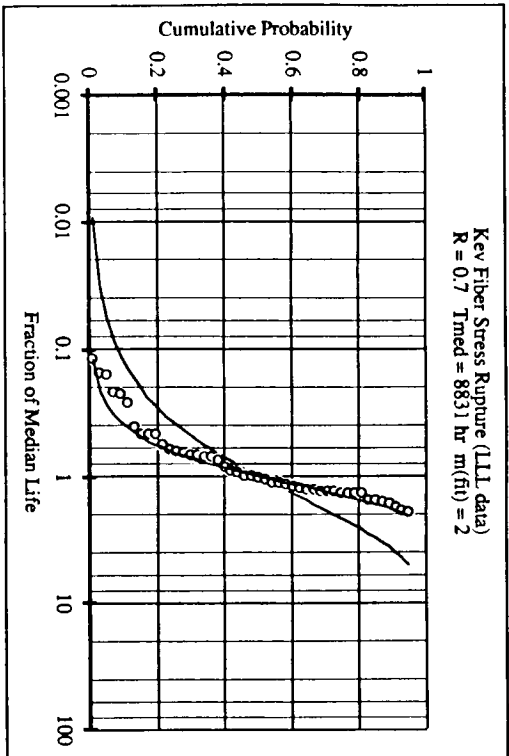


| UCID 19849<br>424.8 ksi 0.84<br>N = 103<br>Med = 12.52 |            |                  |                |         |       |            |                                                        |        |            |                  |                |       |       |                       |
|--------------------------------------------------------|------------|------------------|----------------|---------|-------|------------|--------------------------------------------------------|--------|------------|------------------|----------------|-------|-------|-----------------------|
| r                                                      | Life - hrs | t <sub>med</sub> | F = (-r-1/2)/N | S = 1-F | b est | b=0.9 pred | UCID 19849<br>424.8 ksi 0.84<br>N = 103<br>Med = 12.52 | r      | Life - hrs | t <sub>med</sub> | F = (-r-1/2)/N | 1-F   | b est | pred t <sub>med</sub> |
| 1                                                      | 0.25       | 0.020            | 0.005          | 0.995   | 1.27  | 0.004      | UCID 19849<br>424.8 ksi 0.84<br>N = 103<br>Med = 12.52 | 51     | 11.13      | 0.889            | 0.490          | 0.510 |       | 0.969                 |
| 2                                                      | 0.31       | 0.025            | 0.015          | 0.985   | 1.04  | 0.014      |                                                        | 52     | 12.52      | 1.000            | 0.500          | 0.500 |       | 1.000                 |
| 3                                                      | 0.44       | 0.035            | 0.024          | 0.976   | 1.00  | 0.024      |                                                        | 53     | 12.94      | 1.034            | 0.510          | 0.490 |       | 1.031                 |
| 4                                                      | 0.45       | 0.036            | 0.034          | 0.966   | 0.90  | 0.036      |                                                        | 54     | 16.42      | 1.312            | 0.519          | 0.481 |       | 1.064                 |
| 5                                                      | 0.57       | 0.046            | 0.044          | 0.956   | 0.89  | 0.048      |                                                        | 55     | 16.47      | 1.315            | 0.529          | 0.471 |       | 1.097                 |
| 6                                                      | 0.65       | 0.052            | 0.053          | 0.947   | 0.86  | 0.060      |                                                        | 56     | 17.68      | 1.412            | 0.539          | 0.461 |       | 1.130                 |
| 7                                                      | 0.73       | 0.058            | 0.063          | 0.937   | 0.83  | 0.072      |                                                        | 57     | 17.79      | 1.421            | 0.549          | 0.451 |       | 1.165                 |
| 8                                                      | 0.89       | 0.071            | 0.073          | 0.927   | 0.84  | 0.085      |                                                        | 58     | 18.45      | 1.474            | 0.558          | 0.442 |       | 1.200                 |
| 9                                                      | 1.06       | 0.085            | 0.083          | 0.917   | 0.84  | 0.099      |                                                        | 59     | 18.95      | 1.514            | 0.568          | 0.432 |       | 1.237                 |
| 10                                                     | 1.22       | 0.097            | 0.092          | 0.908   | 0.85  | 0.112      |                                                        | 60     | 19.07      | 1.523            | 0.578          | 0.422 |       | 1.274                 |
| 11                                                     | 1.37       | 0.109            | 0.102          | 0.898   | 0.84  | 0.126      |                                                        | 61     | 19.40      | 1.550            | 0.587          | 0.413 |       | 1.312                 |
| 12                                                     | 1.83       | 0.146            | 0.112          | 0.888   | 0.92  | 0.140      |                                                        | 62     | 19.62      | 1.567            | 0.597          | 0.403 |       | 1.352                 |
| 13                                                     | 1.96       | 0.157            | 0.121          | 0.879   | 0.91  | 0.155      |                                                        | 63     | 19.86      | 1.586            | 0.607          | 0.393 |       | 1.392                 |
| 14                                                     | 2.15       | 0.172            | 0.131          | 0.869   | 0.91  | 0.170      |                                                        | 64     | 20.76      | 1.658            | 0.617          | 0.383 |       | 1.433                 |
| 15                                                     | 2.40       | 0.192            | 0.141          | 0.859   | 0.92  | 0.185      |                                                        | 65     | 21.38      | 1.708            | 0.626          | 0.374 |       | 1.476                 |
| 16                                                     | 2.51       | 0.200            | 0.150          | 0.850   | 0.90  | 0.200      |                                                        | 66     | 23.03      | 1.839            | 0.636          | 0.364 |       | 1.520                 |
| 17                                                     | 2.77       | 0.221            | 0.160          | 0.840   | 0.91  | 0.216      |                                                        | 67     | 23.10      | 1.845            | 0.646          | 0.354 | 0.66  | 1.565                 |
| 18                                                     | 4.05       | 0.323            | 0.170          | 0.830   | 1.16  | 0.232      |                                                        | 68     | 23.83      | 1.903            | 0.655          | 0.345 | 0.67  | 1.612                 |
| 19                                                     | 4.07       | 0.325            | 0.180          | 0.820   | 1.12  | 0.248      |                                                        | 69     | 24.46      | 1.954            | 0.665          | 0.335 | 0.68  | 1.660                 |
| 20                                                     | 4.34       | 0.347            | 0.189          | 0.811   | 1.13  | 0.265      |                                                        | 70     | 24.81      | 1.982            | 0.675          | 0.325 | 0.71  | 1.710                 |
| 21                                                     | 4.98       | 0.398            | 0.199          | 0.801   | 1.24  | 0.282      |                                                        | 71     | 25.15      | 2.009            | 0.684          | 0.316 | 0.73  | 1.761                 |
| 22                                                     | 5.86       | 0.468            | 0.209          | 0.791   | 1.43  | 0.299      |                                                        | 72     | 25.18      | 2.011            | 0.694          | 0.306 | 0.77  | 1.814                 |
| 23                                                     | 5.90       | 0.471            | 0.218          | 0.782   | 1.37  | 0.317      |                                                        | 73     | 26.96      | 2.153            | 0.704          | 0.296 | 0.73  | 1.869                 |
| 24                                                     | 6.18       | 0.494            | 0.228          | 0.772   | 1.39  | 0.335      |                                                        | 74     | 27.53      | 2.199            | 0.714          | 0.286 | 0.75  | 1.926                 |
| 25                                                     | 6.30       | 0.503            | 0.238          | 0.762   | 1.36  | 0.353      |                                                        | 75     | 27.86      | 2.225            | 0.723          | 0.277 | 0.77  | 1.985                 |
| 26                                                     | 7.14       | 0.570            | 0.248          | 0.752   | 1.59  | 0.372      |                                                        | 76     | 29.89      | 2.387            | 0.733          | 0.267 | 0.74  | 2.047                 |
| 27                                                     | 7.32       | 0.585            | 0.257          | 0.743   | 1.58  | 0.391      |                                                        | 77     | 32.55      | 2.600            | 0.743          | 0.257 | 0.70  | 2.110                 |
| 28                                                     | 7.45       | 0.595            | 0.267          | 0.733   | 1.55  | 0.410      |                                                        | 78     | 33.95      | 2.712            | 0.752          | 0.248 | 0.70  | 2.177                 |
| 29                                                     | 7.92       | 0.633            | 0.277          | 0.723   | 1.66  | 0.429      |                                                        | 79     | 34.51      | 2.756            | 0.762          | 0.238 | 0.72  | 2.246                 |
| 30                                                     | 7.99       | 0.638            | 0.286          | 0.714   | 1.60  | 0.449      |                                                        | 80     | 34.74      | 2.775            | 0.772          | 0.228 | 0.74  | 2.319                 |
| 31                                                     | 8.22       | 0.657            | 0.296          | 0.704   | 1.62  | 0.470      |                                                        | 81     | 35.63      | 2.846            | 0.782          | 0.218 | 0.75  | 2.395                 |
| 32                                                     | 8.35       | 0.667            | 0.306          | 0.694   | 1.58  | 0.490      |                                                        | 82     | 36.84      | 2.942            | 0.791          | 0.209 | 0.76  | 2.475                 |
| 33                                                     | 8.38       | 0.669            | 0.316          | 0.684   | 1.50  | 0.511      |                                                        | 83     | 36.97      | 2.953            | 0.801          | 0.199 | 0.78  | 2.558                 |
| 34                                                     | 8.39       | 0.670            | 0.325          | 0.675   | 1.42  | 0.533      |                                                        | 84     | 39.17      | 3.129            | 0.811          | 0.189 | 0.77  | 2.647                 |
| 35                                                     | 8.45       | 0.675            | 0.335          | 0.665   | 1.35  | 0.555      |                                                        | 85     | 41.54      | 3.318            | 0.820          | 0.180 | 0.76  | 2.740                 |
| 36                                                     | 8.53       | 0.681            | 0.345          | 0.655   | 1.29  | 0.577      |                                                        | 86     | 42.32      | 3.380            | 0.830          | 0.170 | 0.77  | 2.838                 |
| 37                                                     | 8.55       | 0.683            | 0.354          | 0.646   | 1.21  | 0.600      |                                                        | 87     | 42.53      | 3.397            | 0.840          | 0.160 | 0.79  | 2.943                 |
| 38                                                     | 8.64       | 0.690            | 0.364          | 0.636   | 1.15  | 0.623      |                                                        | 88     | 43.61      | 3.483            | 0.850          | 0.150 | 0.81  | 3.055                 |
| 39                                                     | 8.68       | 0.693            | 0.374          | 0.626   | 1.07  | 0.646      |                                                        | 89     | 48.54      | 3.877            | 0.859          | 0.141 | 0.77  | 3.175                 |
| 40                                                     | 8.92       | 0.712            | 0.383          | 0.617   | 1.06  | 0.670      |                                                        | 90     | 49.02      | 3.915            | 0.869          | 0.131 | 0.79  | 3.304                 |
| 41                                                     | 8.93       | 0.713            | 0.393          | 0.607   | 0.97  | 0.695      |                                                        | 91     | 55.20      | 4.409            | 0.879          | 0.121 | 0.75  | 3.443                 |
| 42                                                     | 9.45       | 0.755            | 0.403          | 0.597   | 1.05  | 0.720      |                                                        | 92     | 55.99      | 4.472            | 0.888          | 0.112 | 0.77  | 3.595                 |
| 43                                                     | 9.57       | 0.764            | 0.413          | 0.587   | 0.98  | 0.745      |                                                        | 93     | 61.37      | 4.902            | 0.898          | 0.102 | 0.75  | 3.761                 |
| 44                                                     | 9.80       | 0.783            | 0.422          | 0.578   | 0.95  | 0.771      |                                                        | 94     | 63.17      | 5.046            | 0.908          | 0.092 | 0.76  | 3.944                 |
| 45                                                     | 9.83       | 0.785            | 0.432          | 0.568   | 0.84  | 0.798      |                                                        | 95     | 66.48      | 5.310            | 0.917          | 0.083 | 0.77  | 4.149                 |
| 46                                                     | 10.60      | 0.847            | 0.442          | 0.558   | 1.04  | 0.825      |                                                        | 96     | 67.06      | 5.356            | 0.927          | 0.073 | 0.79  | 4.381                 |
| 47                                                     | 10.82      | 0.864            | 0.451          | 0.549   | 0.98  | 0.853      |                                                        | 97     | 74.01      | 5.911            | 0.937          | 0.063 | 0.78  | 4.648                 |
| 48                                                     | 10.83      | 0.865            | 0.461          | 0.539   | 0.79  | 0.881      |                                                        | 98     | 74.61      | 5.959            | 0.947          | 0.053 | 0.81  | 4.961                 |
| 49                                                     | 11.03      | 0.881            | 0.471          | 0.529   | 0.67  | 0.910      |                                                        | 99     | 76.46      | 6.107            | 0.956          | 0.044 | 0.83  | 5.340                 |
| 50                                                     | 11.12      | 0.888            | 0.481          | 0.519   |       | 0.939      |                                                        | 100    | 84.26      | 6.730            | 0.966          | 0.034 | 0.83  | 5.819                 |
|                                                        |            |                  |                |         |       |            | 101                                                    | 89.87  | 7.178      | 0.976            | 0.024          | 0.85  | 6.465 |                       |
|                                                        |            |                  |                |         |       |            | 102                                                    | 97.37  | 7.777      | 0.985            | 0.015          | 0.88  | 7.460 |                       |
|                                                        |            |                  |                |         |       |            | 103                                                    | 119.09 | 9.512      | 0.995            | 0.005          | 0.91  | 9.641 |                       |
| more                                                   |            |                  |                |         |       |            |                                                        |        |            |                  |                |       |       |                       |

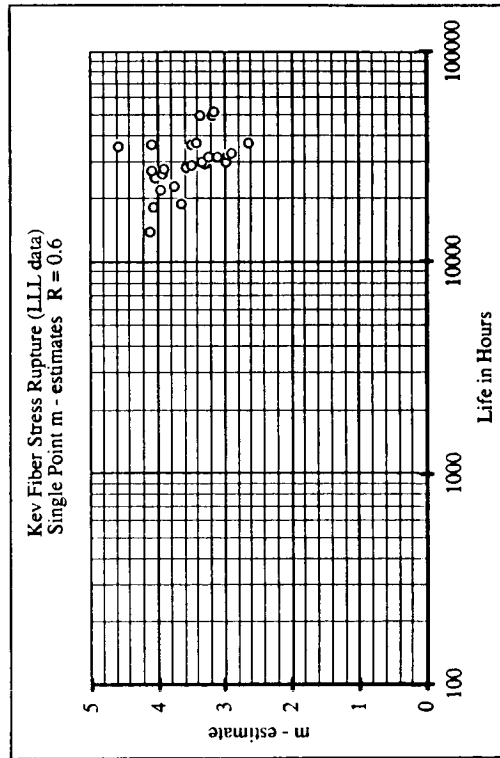
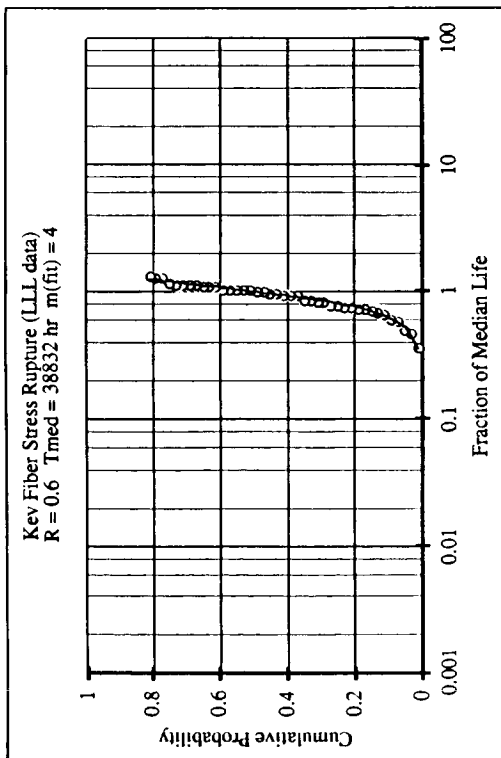
more



| UCID 19849     |            |       |               |         |       |             |  |  |  | UCID 19849     |            |       |               |         |       |            |  |  |  |
|----------------|------------|-------|---------------|---------|-------|-------------|--|--|--|----------------|------------|-------|---------------|---------|-------|------------|--|--|--|
| 404.6 ksi 0.8  |            |       |               |         |       |             |  |  |  | 404.6 ksi 0.8  |            |       |               |         |       |            |  |  |  |
| N = 100        |            |       |               |         |       |             |  |  |  | N = 100        |            |       |               |         |       |            |  |  |  |
| Med = 150.7 hr |            |       |               |         |       |             |  |  |  | Med = 150.7 hr |            |       |               |         |       |            |  |  |  |
| r              | Life - hrs | Wmed  | F = (r-1/2)/N | S = 1-F | m est | m = 0.9 fit |  |  |  | r              | Life - hrs | Wmed  | F = (r-1/2)/N | S = 1-F | b est | b=0.9 pred |  |  |  |
| 1              | 1.80       | 0.012 | 0.005         | 0.995   | 1.11  | 0.004       |  |  |  | 51             | 152.20     | 1.010 | 0.505         | 0.495   |       | 1.016      |  |  |  |
| 2              | 3.10       | 0.021 | 0.015         | 0.985   | 0.99  | 0.014       |  |  |  | 52             | 152.80     | 1.014 | 0.515         | 0.485   |       | 1.049      |  |  |  |
| 3              | 4.20       | 0.028 | 0.025         | 0.975   | 0.92  | 0.025       |  |  |  | 53             | 157.70     | 1.046 | 0.525         | 0.475   | 1.57  | 1.083      |  |  |  |
| 4              | 6.00       | 0.040 | 0.035         | 0.965   | 0.92  | 0.037       |  |  |  | 54             | 160.00     | 1.062 | 0.535         | 0.465   | 1.66  | 1.117      |  |  |  |
| 5              | 7.50       | 0.050 | 0.045         | 0.955   | 0.90  | 0.049       |  |  |  | 55             | 163.60     | 1.086 | 0.545         | 0.455   | 1.55  | 1.152      |  |  |  |
| 6              | 8.20       | 0.054 | 0.055         | 0.945   | 0.86  | 0.062       |  |  |  | 56             | 166.90     | 1.107 | 0.555         | 0.445   | 1.52  | 1.188      |  |  |  |
| 7              | 8.50       | 0.056 | 0.065         | 0.935   | 0.81  | 0.075       |  |  |  | 57             | 170.50     | 1.131 | 0.565         | 0.435   | 1.48  | 1.226      |  |  |  |
| 8              | 10.30      | 0.068 | 0.075         | 0.925   | 0.81  | 0.088       |  |  |  | 58             | 174.90     | 1.161 | 0.575         | 0.425   | 1.41  | 1.264      |  |  |  |
| 9              | 10.60      | 0.070 | 0.085         | 0.915   | 1.06  | 0.102       |  |  |  | 59             | 177.70     | 1.179 | 0.585         | 0.415   | 1.44  | 1.303      |  |  |  |
| 10             | 24.20      | 0.161 | 0.095         | 0.905   | 1.13  | 0.131       |  |  |  | 60             | 179.20     | 1.189 | 0.595         | 0.405   | 1.53  | 1.343      |  |  |  |
| 11             | 29.60      | 0.196 | 0.105         | 0.895   | 1.11  | 0.145       |  |  |  | 61             | 183.60     | 1.218 | 0.605         | 0.395   | 1.48  | 1.384      |  |  |  |
| 12             | 31.70      | 0.210 | 0.115         | 0.885   | 1.29  | 0.160       |  |  |  | 62             | 183.80     | 1.220 | 0.615         | 0.385   | 1.61  | 1.427      |  |  |  |
| 13             | 41.90      | 0.278 | 0.125         | 0.875   | 1.27  | 0.176       |  |  |  | 63             | 194.30     | 1.289 | 0.625         | 0.375   | 1.37  | 1.471      |  |  |  |
| 14             | 44.10      | 0.293 | 0.135         | 0.865   | 1.34  | 0.192       |  |  |  | 64             | 195.10     | 1.295 | 0.635         | 0.365   | 1.45  | 1.516      |  |  |  |
| 15             | 49.50      | 0.328 | 0.145         | 0.855   | 1.28  | 0.208       |  |  |  | 65             | 195.30     | 1.296 | 0.645         | 0.355   | 1.55  | 1.562      |  |  |  |
| 16             | 50.10      | 0.332 | 0.155         | 0.845   | 1.45  | 0.224       |  |  |  | 66             | 202.60     | 1.344 | 0.655         | 0.345   | 1.45  | 1.610      |  |  |  |
| 17             | 59.70      | 0.396 | 0.165         | 0.835   | 1.44  | 0.258       |  |  |  | 67             | 220.20     | 1.461 | 0.665         | 0.335   | 1.20  | 1.660      |  |  |  |
| 18             | 61.70      | 0.409 | 0.175         | 0.825   | 1.51  | 0.275       |  |  |  | 68             | 227.20     | 1.508 | 0.675         | 0.325   | 1.26  | 1.711      |  |  |  |
| 19             | 64.40      | 0.427 | 0.185         | 0.815   | 1.44  | 0.293       |  |  |  | 69             | 231.00     | 1.666 | 0.685         | 0.315   | 1.24  | 1.764      |  |  |  |
| 20             | 69.70      | 0.463 | 0.195         | 0.805   | 1.59  | 0.311       |  |  |  | 70             | 251.00     | 1.666 | 0.695         | 0.305   | 1.06  | 1.819      |  |  |  |
| 21             | 70.00      | 0.464 | 0.205         | 0.795   | 1.44  | 0.329       |  |  |  | 71             | 266.50     | 1.768 | 0.705         | 0.295   | 0.99  | 1.876      |  |  |  |
| 22             | 77.80      | 0.516 | 0.215         | 0.785   | 1.60  | 0.348       |  |  |  | 72             | 267.90     | 1.778 | 0.715         | 0.285   | 1.03  | 1.934      |  |  |  |
| 23             | 80.50      | 0.534 | 0.225         | 0.775   | 1.57  | 0.367       |  |  |  | 73             | 269.20     | 1.786 | 0.725         | 0.275   | 1.07  | 1.996      |  |  |  |
| 24             | 82.30      | 0.546 | 0.235         | 0.765   | 1.53  | 0.386       |  |  |  | 74             | 270.40     | 1.794 | 0.735         | 0.265   | 1.11  | 2.059      |  |  |  |
| 25             | 83.50      | 0.554 | 0.245         | 0.755   | 1.47  | 0.406       |  |  |  | 75             | 272.50     | 1.808 | 0.745         | 0.255   | 1.15  | 2.126      |  |  |  |
| 26             | 84.20      | 0.559 | 0.255         | 0.745   | 1.48  | 0.426       |  |  |  | 76             | 285.90     | 1.897 | 0.755         | 0.245   | 1.11  | 2.195      |  |  |  |
| 27             | 87.10      | 0.578 | 0.265         | 0.735   | 1.41  | 0.446       |  |  |  | 77             | 292.60     | 1.942 | 0.765         | 0.235   | 1.11  | 2.268      |  |  |  |
| 28             | 87.30      | 0.579 | 0.275         | 0.725   | 1.51  | 0.467       |  |  |  | 78             | 295.10     | 1.958 | 0.775         | 0.225   | 1.14  | 2.343      |  |  |  |
| 29             | 93.20      | 0.618 | 0.285         | 0.715   | 1.82  | 0.489       |  |  |  | 79             | 301.10     | 1.998 | 0.785         | 0.215   | 1.15  | 2.423      |  |  |  |
| 30             | 103.40     | 0.686 | 0.295         | 0.705   | 1.77  | 0.510       |  |  |  | 80             | 304.30     | 2.019 | 0.795         | 0.205   | 1.18  | 2.506      |  |  |  |
| 31             | 104.60     | 0.694 | 0.305         | 0.695   | 1.70  | 0.532       |  |  |  | 81             | 316.80     | 2.102 | 0.805         | 0.195   | 1.15  | 2.594      |  |  |  |
| 32             | 105.50     | 0.700 | 0.315         | 0.685   | 1.82  | 0.555       |  |  |  | 82             | 329.80     | 2.188 | 0.815         | 0.185   | 1.14  | 2.687      |  |  |  |
| 33             | 108.80     | 0.722 | 0.325         | 0.675   | 1.82  | 0.578       |  |  |  | 83             | 334.10     | 2.217 | 0.825         | 0.175   | 1.16  | 2.786      |  |  |  |
| 34             | 112.60     | 0.747 | 0.335         | 0.665   | 1.94  | 0.601       |  |  |  | 84             | 346.20     | 2.297 | 0.835         | 0.165   | 1.15  | 2.891      |  |  |  |
| 35             | 116.80     | 0.775 | 0.345         | 0.655   | 2.00  | 0.625       |  |  |  | 85             | 351.20     | 2.330 | 0.845         | 0.155   | 1.17  | 3.002      |  |  |  |
| 36             | 118.00     | 0.783 | 0.355         | 0.645   | 1.85  | 0.649       |  |  |  | 86             | 353.30     | 2.344 | 0.855         | 0.145   | 1.20  | 3.122      |  |  |  |
| 37             | 122.00     | 0.810 | 0.365         | 0.635   | 2.00  | 0.674       |  |  |  | 87             | 369.30     | 2.451 | 0.865         | 0.135   | 1.18  | 3.250      |  |  |  |
| 38             | 123.50     | 0.820 | 0.375         | 0.625   | 1.95  | 0.700       |  |  |  | 88             | 372.30     | 2.470 | 0.875         | 0.125   | 1.21  | 3.389      |  |  |  |
| 39             | 124.40     | 0.825 | 0.385         | 0.615   | 1.75  | 0.725       |  |  |  | 89             | 381.30     | 2.530 | 0.885         | 0.115   | 1.23  | 3.541      |  |  |  |
| 40             | 125.40     | 0.832 | 0.395         | 0.605   | 1.91  | 0.752       |  |  |  | 90             | 393.50     | 2.611 | 0.895         | 0.105   | 1.23  | 3.707      |  |  |  |
| 41             | 129.50     | 0.859 | 0.405         | 0.595   | 1.78  | 0.779       |  |  |  | 91             | 451.30     | 2.995 | 0.905         | 0.095   | 1.11  | 3.890      |  |  |  |
| 42             | 130.40     | 0.865 | 0.415         | 0.585   | 1.82  | 0.806       |  |  |  | 92             | 461.50     | 3.062 | 0.915         | 0.085   | 1.13  | 4.095      |  |  |  |
| 43             | 131.60     | 0.873 | 0.425         | 0.575   | 1.53  | 0.834       |  |  |  | 93             | 574.20     | 3.810 | 0.925         | 0.075   | 0.99  | 4.326      |  |  |  |
| 44             | 132.80     | 0.881 | 0.435         | 0.565   | 1.39  | 0.863       |  |  |  | 94             | 653.00     | 4.335 | 0.935         | 0.065   | 0.94  | 4.593      |  |  |  |
| 45             | 133.80     | 0.888 | 0.445         | 0.555   | 1.42  | 0.892       |  |  |  | 95             | 663.00     | 4.399 | 0.945         | 0.055   | 0.97  | 4.906      |  |  |  |
| 46             | 137.00     | 0.909 | 0.455         | 0.545   | 1.09  | 0.922       |  |  |  | 96             | 669.80     | 4.445 | 0.955         | 0.045   | 1.00  | 5.284      |  |  |  |
| 47             | 140.20     | 0.930 | 0.465         | 0.535   | 0.953 | 0.953       |  |  |  | 97             | 739.70     | 4.908 | 0.965         | 0.035   | 0.99  | 5.762      |  |  |  |
| 48             | 140.90     | 0.935 | 0.475         | 0.525   | 0.984 |             |  |  |  | 98             | 759.60     | 5.040 | 0.975         | 0.025   | 1.03  | 6.408      |  |  |  |
| 49             | 148.50     | 0.985 | 0.485         | 0.515   |       |             |  |  |  | 99             | 894.70     | 5.937 | 0.985         | 0.015   | 1.01  | 7.402      |  |  |  |
| 50             | 149.20     | 0.990 | 0.495         | 0.505   |       |             |  |  |  | 100            | 974.90     | 6.469 | 0.995         | 0.005   | 1.09  | 9.582      |  |  |  |

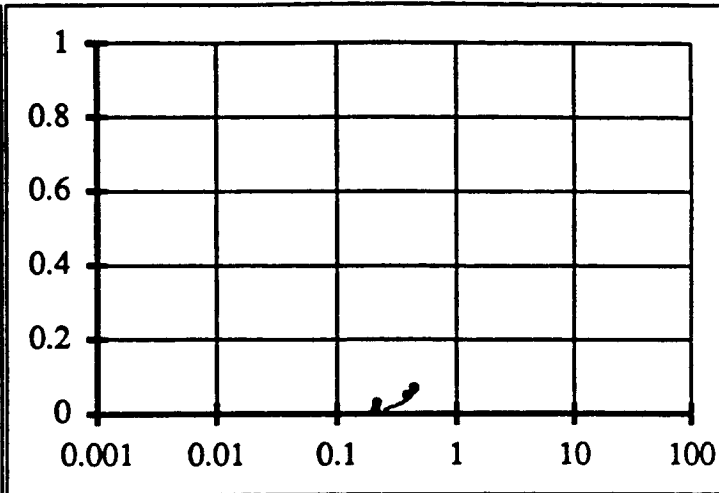
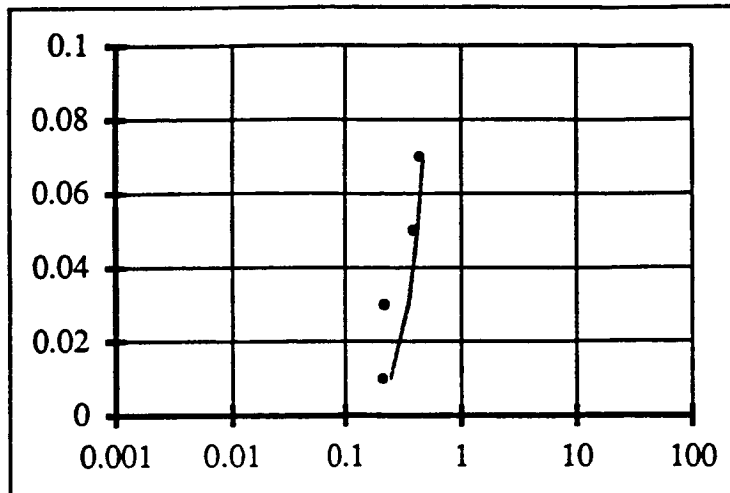


| UCID19849     | 1  | Life_hrs | Wound | $F = (r-1/2)/N$ | $S = 1-F$ | W ent | $m = 0.9$ fit | $m = 1.5$ | $m = 2$ |
|---------------|----|----------|-------|-----------------|-----------|-------|---------------|-----------|---------|
| 354 km 0.7    | 2  | 1051     | 0.119 | 0.010           | 0.990     | 1.98  | 0.009         | 0.060     | 0.122   |
| N = 49        | 3  | 1389     | 0.151 | 0.031           | 0.969     | 1.64  | 0.032         | 0.169     | 0.212   |
| Med = 8831 hr | 4  | 1921     | 0.157 | 0.051           | 0.949     | 1.40  | 0.057         | 0.158     | 0.215   |
|               | 5  | 1942     | 0.218 | 0.071           | 0.929     | 1.47  | 0.083         | 0.203     | 0.237   |
|               | 6  | 2322     | 0.263 | 0.112           | 0.888     | 1.32  | 0.112         | 0.244     | 0.273   |
|               | 7  | 3629     | 0.411 | 0.133           | 0.867     | 1.78  | 0.172         | 0.323     | 0.453   |
|               | 8  | 4006     | 0.454 | 0.153           | 0.847     | 1.81  | 0.204         | 0.360     | 0.490   |
|               | 9  | 4012     | 0.454 | 0.173           | 0.827     | 1.64  | 0.238         | 0.398     | 0.524   |
|               | 10 | 4921     | 0.460 | 0.194           | 0.806     | 1.50  | 0.273         | 0.434     | 0.558   |
|               | 11 | 4921     | 0.557 | 0.214           | 0.786     | 1.81  | 0.309         | 0.470     | 0.590   |
|               | 12 | 5445     | 0.617 | 0.235           | 0.765     | 1.97  | 0.347         | 0.507     | 0.621   |
|               | 13 | 5620     | 0.636 | 0.255           | 0.745     | 1.89  | 0.386         | 0.543     | 0.652   |
|               | 14 | 5817     | 0.659 | 0.276           | 0.724     | 1.83  | 0.427         | 0.579     | 0.682   |
|               | 15 | 5905     | 0.669 | 0.296           | 0.704     | 1.69  | 0.469         | 0.615     | 0.711   |
|               | 16 | 5956     | 0.674 | 0.316           | 0.684     | 1.52  | 0.513         | 0.651     | 0.741   |
|               | 17 | 6068     | 0.687 | 0.337           | 0.663     | 1.40  | 0.559         | 0.688     | 0.770   |
|               | 18 | 6121     | 0.693 | 0.357           | 0.643     | 1.23  | 0.606         | 0.725     | 0.798   |
|               | 19 | 6473     | 0.733 | 0.378           | 0.622     | 1.22  | 0.656         | 0.762     | 0.827   |
|               | 20 | 7501     | 0.849 | 0.398           | 0.602     | 1.91  | 0.707         | 0.800     | 0.856   |
|               | 21 | 7866     | 0.893 | 0.418           | 0.582     | 2.17  | 0.761         | 0.839     | 0.884   |
|               | 22 | 8105     | 0.918 | 0.439           | 0.561     | 2.13  | 0.817         | 0.878     | 0.913   |
|               | 23 | 8546     | 0.968 | 0.459           | 0.541     |       | 0.875         | 0.918     | 0.942   |
|               | 24 | 8666     | 0.981 | 0.480           | 0.520     |       | 0.936         | 0.958     | 0.971   |
|               | 25 | 8831     | 1.000 | 0.500           | 0.500     | 1.90  | 1.000         | 1.000     | 1.000   |
|               | 26 | 9106     | 1.031 | 0.520           | 0.480     | 1.67  | 1.067         | 1.043     | 1.030   |
|               | 27 | 9711     | 1.100 | 0.541           | 0.459     | 1.22  | 1.137         | 1.086     | 1.060   |
|               | 28 | 9806     | 1.110 | 0.561           | 0.439     | 1.65  | 1.211         | 1.131     | 1.090   |
|               | 29 | 10205    | 1.156 | 0.582           | 0.418     | 1.58  | 1.290         | 1.178     | 1.121   |
|               | 30 | 10396    | 1.177 | 0.602           | 0.398     | 1.74  | 1.372         | 1.225     | 1.153   |
|               | 31 | 10861    | 1.230 | 0.622           | 0.378     | 1.64  | 1.459         | 1.275     | 1.185   |
|               | 32 | 11026    | 1.249 | 0.643           | 0.357     | 1.78  | 1.552         | 1.327     | 1.219   |
|               | 33 | 11214    | 1.270 | 0.663           | 0.337     | 1.89  | 1.651         | 1.380     | 1.253   |
|               | 34 | 11362    | 1.287 | 0.684           | 0.316     | 2.01  | 1.757         | 1.437     | 1.289   |
|               | 35 | 11604    | 1.314 | 0.704           | 0.296     | 2.06  | 1.870         | 1.496     | 1.325   |
|               | 36 | 11608    | 1.314 | 0.724           | 0.276     | 2.27  | 1.993         | 1.558     | 1.364   |
|               | 37 | 11745    | 1.330 | 0.745           | 0.255     | 2.38  | 2.125         | 1.624     | 1.404   |
|               | 38 | 11762    | 1.332 | 0.765           | 0.235     | 2.57  | 2.270         | 1.694     | 1.446   |
|               | 39 | 11895    | 1.347 | 0.786           | 0.214     | 2.68  | 2.429         | 1.769     | 1.491   |
|               | 40 | 12044    | 1.364 | 0.806           | 0.194     | 2.78  | 2.605         | 1.850     | 1.538   |
|               | 41 | 13520    | 1.531 | 0.827           | 0.173     | 2.18  | 2.801         | 1.939     | 1.590   |
|               | 42 | 13670    | 1.598 | 0.847           | 0.153     | 2.28  | 3.025         | 2.057     | 1.646   |
|               | 43 | 14110    | 1.598 | 0.867           | 0.133     | 2.28  | 3.282         | 2.147     | 1.707   |
|               | 44 | 14496    | 1.641 | 0.888           | 0.112     | 2.32  | 3.585         | 2.272     | 1.766   |
|               | 45 | 15395    | 1.743 | 0.908           | 0.092     | 2.23  | 3.952         | 2.419     | 1.856   |
|               | 46 | 16179    | 1.832 | 0.929           | 0.071     | 2.21  | 4.417         | 2.599     | 1.951   |
|               | 47 | 17902    | 1.935 | 0.949           | 0.051     | 2.21  | 5.047         | 2.831     | 2.072   |



| UCID 19849     | r  | Life - hrs | vmed  | F = (c-12)/N | S = 1-F | b est | b = 4 fit |
|----------------|----|------------|-------|--------------|---------|-------|-----------|
| 303.5 ksi 0.6  | 1  | 13872      | 0.357 | 0.010        | 0.990   | 4.11  | 0.347     |
| N = 50         | 2  | 18024      | 0.464 | 0.030        | 0.970   | 4.07  | 0.458     |
| Med = 38832 hr | 3  | 19008      | 0.489 | 0.050        | 0.950   | 3.64  | 0.522     |
|                | 4  | 21960      | 0.566 | 0.070        | 0.930   | 3.96  | 0.569     |
|                | 5  | 22872      | 0.589 | 0.090        | 0.910   | 3.77  | 0.607     |
|                | 6  | 25008      | 0.644 | 0.110        | 0.890   | 4.05  | 0.640     |
|                | 7  | 25848      | 0.666 | 0.130        | 0.870   | 3.94  | 0.670     |
|                | 8  | 27216      | 0.701 | 0.150        | 0.850   | 4.08  | 0.696     |
|                | 9  | 27744      | 0.714 | 0.170        | 0.830   | 3.91  | 0.720     |
|                | 10 | 27840      | 0.717 | 0.190        | 0.810   | 3.58  | 0.743     |
|                | 11 | 28512      | 0.734 | 0.210        | 0.790   | 3.49  | 0.764     |
|                | 12 | 28896      | 0.744 | 0.230        | 0.770   | 3.30  | 0.784     |
|                | 13 | 29832      | 0.768 | 0.250        | 0.750   | 3.34  | 0.803     |
|                | 14 | 29832      | 0.768 | 0.270        | 0.730   | 2.99  | 0.821     |
|                | 15 | 31224      | 0.804 | 0.290        | 0.710   | 3.23  | 0.838     |
|                | 16 | 31752      | 0.818 | 0.310        | 0.690   | 3.10  | 0.855     |
|                | 17 | 32232      | 0.830 | 0.330        | 0.670   | 2.94  | 0.872     |
|                | 18 | 32976      | 0.849 | 0.350        | 0.650   | 2.91  | 0.888     |
|                | 19 | 35544      | 0.915 | 0.370        | 0.630   | 4.58  | 0.904     |
|                | 20 | 35760      | 0.921 | 0.390        | 0.610   | 4.10  | 0.919     |
|                | 21 | 35928      | 0.925 | 0.410        | 0.590   | 3.51  | 0.934     |
|                | 22 | 36528      | 0.941 | 0.430        | 0.570   | 3.43  | 0.949     |
|                | 23 | 36720      | 0.946 | 0.450        | 0.550   | 2.65  | 0.964     |
|                | 24 | 38592      | 0.994 | 0.470        | 0.530   |       | 0.978     |
| repeat *       | 25 | 38592      | 0.994 | 0.490        | 0.510   |       | 0.993     |
|                | 26 | 39072      | 1.006 | 0.510        | 0.490   |       | 1.007     |
|                | 27 | 39240      | 1.011 | 0.530        | 0.470   |       | 1.022     |
|                | 28 | 39576      | 1.019 | 0.550        | 0.450   |       | 1.036     |
|                | 29 | 39744      | 1.023 | 0.570        | 0.430   |       | 1.050     |
| repeat *       | 30 | 39744      | 1.023 | 0.590        | 0.410   |       | 1.065     |
|                | 31 | 41592      | 1.071 | 0.610        | 0.390   |       | 1.080     |
|                | 32 | 41760      | 1.075 | 0.630        | 0.370   |       | 1.094     |
| repeat *       | 33 | 41760      | 1.075 | 0.650        | 0.350   |       | 1.109     |
|                | 34 | 42600      | 1.097 | 0.670        | 0.330   |       | 1.125     |
| repeat *       | 35 | 42600      | 1.097 | 0.690        | 0.310   |       | 1.140     |
|                | 36 | 42960      | 1.106 | 0.710        | 0.290   |       | 1.156     |
|                | 37 | 43176      | 1.112 | 0.730        | 0.270   |       | 1.172     |
|                | 38 | 44664      | 1.150 | 0.750        | 0.250   |       | 1.189     |
|                | 39 | 49176      | 1.266 | 0.770        | 0.230   | 3.18  | 1.207     |
|                | 40 | 49440      | 1.273 | 0.790        | 0.210   | 3.36  | 1.225     |
|                | 41 | 51192      | 1.318 | 0.810        | 0.190   | 3.16  | 1.244     |

|                  |   |            |       |               |         |       |            |       |       |       |
|------------------|---|------------|-------|---------------|---------|-------|------------|-------|-------|-------|
| UCID 19849       | r | Life - hrs | t/t4  | F = (r-1/2)/N | S = 1-p | m est | Est Median | t/med | m fit | m =   |
| 252 ksi 0.5      | 1 | 31344      | 0.475 | 0.010         | 0.990   | 2.65  | 154536     | 0.209 | 0.244 | 3.000 |
| N = 50           | 2 | 32376      | 0.490 | 0.030         | 0.970   | 1.22  | 420866     | 0.216 | 0.353 |       |
| r=4 life = 66024 | 3 | 58056      | 0.879 | 0.050         | 0.950   | 2.70  | 152386     | 0.387 | 0.420 |       |
|                  | 4 | 66024      | 1.000 | 0.070         | 0.930   |       |            | 0.440 | 0.471 |       |
| (150,000)        |   |            |       |               |         |       |            |       |       |       |



**Designing for Time-Dependent Material Response  
in Spacecraft Structures**

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**ABSTRACT**

To study the influence on overall deformations of the time-dependent constitutive properties of fiber-reinforced polymeric matrix composite materials being considered for use in orbiting precision segmented reflectors, simple sandwich beam models are developed. The beam models include layers representing the face sheets, the core, and the adhesive bonding of the face sheets to the core. A three-layer model lumps the adhesive layers with the face sheets or core, while a five-layer model considers the adhesive layers explicitly. The deformation response of the three-layer and five-layer sandwich beam models to a midspan point load is studied. This elementary loading leads to a simple analysis, and it is easy to create this loading in the laboratory. Using the correspondence principle of viscoelasticity, the models representing the elastic behavior of the two beams are transformed into time-dependent models. Representative cases of time-dependent material behavior for the face-sheet material, the core material, and the adhesive are used to evaluate the influence of these constituents being time-dependent on the deformations of the beam. As an example of the results presented, if it assumed that, as a worst case, the polymer-dominated shear properties of the core behave as a Maxwell fluid such that under constant shear stress the shear strain increases by a factor of 10 in 20 years, then it is shown that the beam deflection increases by a factor of 1.4 during that time. In addition to quantitative conclusions, several assumptions are discussed which simplify the analyses for use with more complicated material models. Finally, it is shown that the simpler three-layer model suffices in many situations.

**INTRODUCTION**

As part of the development phase of the use of composite materials for long-duration space applications, it would be useful to have a simple analytical tool which models the important features of sandwich construction and allows for the evaluation of the influence of the time-dependent behavior of the various constituents in the construction; namely the face sheets, the core, and the adhesive. In addition, it would be beneficial to have a simple laboratory specimen which could be used to gather empirical data regarding time-dependent material behavior and screen candidate materials. Both of these requirements can be satisfied to a

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large degree by considering sandwich beams, and by studying their time-dependent behavior as a function of the time-dependent behavior of the constituents. Beams are one-dimensional in nature, leading to somewhat simpler analyses than plate or shell specimens, and laboratory loading of beams is generally direct and free of unwanted secondary effects. Three- or four-point loadings are examples of simple yet effective loadings. This paper discusses the time-dependent response of both three- and five-layer symmetric sandwich beams. The five-layer model includes the face sheets and the core, and the adhesive bonding these constituents together. The three-layer model lumps the adhesive layers into either the face sheets or the core. The response of these two models to a three-point loading is considered. Numerical predictions regarding the deflections over a 20 year time span are made in the context of the various constituents of the sandwich construction exhibiting time-dependent behavior. Since 20 year data are not available, the behavior of the constituents is hypothesized. For more complete details of the work discussed, the reader should consult ref. 1.

## DEVELOPMENT OF THE THREE-LAYER MODEL

### Nomenclature and Problem Definition

In fig. 1 the three-layer model is described, as is the loading. The beams considered are of length  $L$ . The three-point bending load consists of a simply supported beam with a point load  $P$  at midspan. Because of symmetry about the midspan, this loading is studied here as a cantilever beam with a tip load  $P/2$ , the center of the simply-supported beam being clamped in the analogous cantilever problem. The extensional moduli in the  $x$  direction of the face sheet and core are denoted as  $E_1$  and  $E_2$ , respectively. The shear moduli in the  $x - z$  plane are denoted as  $G_1$  and  $G_2$ . The thickness of the face sheets is  $t_1$  and that of the core  $2h$ . The overall thickness is  $2H$ .

### Equations Governing Elastic Response

For this problem it is assumed that the stress components  $\sigma_y$ ,  $\sigma_z$ ,  $\tau_{xy}$ , and  $\tau_{yz}$  are zero. Hence the elastic stress-strain relation is given by

$$\sigma_x = E \epsilon_x \quad \tau_{xz} = G \gamma_{xz} \quad (1)$$

In the above, for a particular layer,  $E$  is the extensional modulus in the  $x$  direction and  $G$  the shear modulus in the  $x - z$  plane.

The assumed displacement field for the three-layer beam is also illustrated in fig. 1. In particular, the sandwich cross section is assumed to displace uniformly as-a-whole in the  $x$  direction an amount  $u^o(x)$  and downward as-a-whole an amount  $w^o(x)$ . In addition, the cross sections of the face sheet layers are assumed to rotate independently of the cross section of the core; the angles of rotation,  $\psi$  and  $\phi$ , respectively, being defined in the figure. With these kinematics, the displacement field is written as

$$\begin{aligned} u(x,z) = & \begin{aligned} & u^o(x) + h\phi(x) - (z+h)\psi(x) & (-H \leq z \leq -h) \\ & u^o(x) - z\phi(x) & (-h \leq z \leq +h) \\ & u^o(x) - h\phi(x) - (z-h)\psi(x) & (+h \leq z \leq +H) \end{aligned} \\ w(x,z) = & w^o(x), \text{ all } z \end{aligned} \quad (2)$$

Hereafter, for convenience, the superscript  $o$  will be dropped and hence the strains required for use in the stress-strain relation are given by

$$\varepsilon_x = \begin{aligned} & \frac{du}{dx} + h \frac{d\phi}{dx} - (z+h) \frac{d\psi}{dx} & \gamma_{xz} = & -\psi + \frac{dw}{dx} & (-H \leq z \leq -h) \\ & \frac{du}{dx} - z \frac{d\phi}{dx} & & -\phi + \frac{dw}{dx} & (-h \leq z \leq +h) \\ & \frac{du}{dx} - h \frac{d\phi}{dx} - (z-h) \frac{d\psi}{dx} & & -\psi + \frac{dw}{dx} & (+h \leq z \leq +H) . \end{aligned} \quad (3)$$

To be determined are  $u(x)$ ,  $\phi(x)$ ,  $\psi(x)$ , and  $w(x)$ . For the viscoelastic problem the functions are time-dependent and should be written as  $u(x,t)$ , ....

The total potential energy used for determining the elastic response simplifies here to

$$\Pi = \frac{1}{2} \int_0^{\frac{L}{2}} \int_{-\frac{H}{2}}^{\frac{H}{2}} (E \varepsilon_x^2 + G \gamma_{xz}^2) dx dz - \frac{P}{2} w\left(\frac{L}{2}\right), \quad (4)$$

where the limits on the integral reflect the fact that the analog cantilever problem is being considered, the tip being loaded with known load  $P/2$ . Substituting the moduli and strains into eq. 4 and integrating with respect to  $z$  leads to

$$\begin{aligned} \Pi = \int_0^{\frac{L}{2}} & \left[ c_0 \left( \frac{du}{dx} \right)^2 + c_1 \left( \frac{d\phi}{dx} \right)^2 + c_3 \left( \frac{d\phi}{dx} \right) \left( \frac{d\psi}{dx} \right) \right. \\ & + c_6 \left( \frac{d\psi}{dx} \right)^2 + \frac{1}{2} (c_7 + c_9) \left( \frac{dw}{dx} \right)^2 \\ & + \frac{1}{2} c_7 \psi^2 + \frac{1}{2} c_9 \phi^2 - c_7 \psi \left( \frac{dw}{dx} \right) - c_9 \phi \left( \frac{dw}{dx} \right) \Big] dx \\ & - \frac{P}{2} w\left(\frac{L}{2}\right). \end{aligned} \quad (5)$$

The constants  $c_i$  are as follows:

$$\begin{aligned} c_0 &= E_1 t_1 + E_2 h \\ c_1 &= h^2 \left( E_1 t_1 + \frac{1}{3} E_2 h \right) \\ c_3 &= E_1 h t_1^2 \\ c_6 &= \frac{1}{3} E_1 t_1^3 \\ c_7 &= 2G_1 t_1 \\ c_9 &= 2G_2 h . \end{aligned} \quad (6)$$

From eq. 5, by taking the first variation and integrating by parts, the governing equations can be shown to be

$$\begin{aligned}
2c_0 \left( \frac{d^2 u}{dx^2} \right) &= 0 \\
2c_1 \left( \frac{d^2 \phi}{dx^2} \right) + c_3 \left( \frac{d^2 \psi}{dx^2} \right) + c_9 \left( \frac{dw}{dx} - \phi \right) &= 0 \\
c_3 \left( \frac{d^2 \phi}{dx^2} \right) + 2c_6 \left( \frac{d^2 \psi}{dx^2} \right) + c_7 \left( \frac{dw}{dx} - \psi \right) &= 0 \\
c_7 \left( \frac{d\psi}{dx} - \frac{d^2 w}{dx^2} \right) + c_9 \left( \frac{d\phi}{dx} - \frac{d^2 w}{dx^2} \right) &= 0 .
\end{aligned} \tag{7}$$

The boundary conditions associated with the variational statement are

$$\begin{array}{ll}
\text{at } x=0 & \text{at } x = \frac{L}{2} \\
u=0 & 2c_0 \frac{du}{dx} = 0 \\
\phi=0 & 2c_1 \frac{d\phi}{dx} + c_3 \frac{d\psi}{dx} = 0 \\
\psi=0 & c_3 \frac{d\phi}{dx} + 2c_6 \frac{d\psi}{dx} = 0 \\
w=0 & (c_7 + c_9) \frac{dw}{dx} - c_7 \psi - c_9 \phi = \frac{P}{2} .
\end{array} \tag{8}$$

On the right side, the 1st and 4th terms can be identified with inplane and shear force resultants, respectively, while the 2nd and 3rd terms are moment resultants.

The equation for  $u(x)$  is decoupled from the other three equations. Here attention will be focused on the displacement  $w(x)$  and thus the first equation will not be considered further. The solutions for the other three displacement variables are taken to be of the form

$$\begin{aligned}
\phi(x) &= fe^{\lambda x} \\
\psi(x) &= se^{\lambda x} \\
w(x) &= we^{\lambda x}
\end{aligned} \tag{9}$$

Substituting these forms into the last three of eq. 7 results in the following polynomial that must be satisfied by  $\lambda$ :

$$\lambda^6 (4c_1 c_6 - c_3^2) (c_7 + c_9) - \lambda^4 (2c_7 c_9) (c_1 + c_3 + c_6) = 0 . \tag{10}$$

The roots to this equation, and application of the boundary conditions, lead to

$$\begin{aligned}
w(x) &= w_3 x^3 + w_2 x^2 + w_1 x + w_0 + w_5 \sinh(\lambda x) + w_6 \cosh(\lambda x) \\
\phi(x) &= 3w_3 x^2 + 2w_2 x + \left[ w_1 + \frac{6(2c_1 + c_3)}{c_9} w_3 \right] + w_5 A_6 \cosh(\lambda x) + w_6 A_6 \sinh(\lambda x) \\
\psi(x) &= 3w_3 x^2 + 2w_2 x + \left[ w_1 + \frac{6(c_3 + 2c_6)}{c_7} w_3 \right] + w_5 B_6 \cosh(\lambda x) + w_6 B_6 \sinh(\lambda x)
\end{aligned} \tag{11}$$

where

$$\begin{aligned}
w_0 &= w_5 \\
w_1 &= \frac{P}{2(R_1 + R_2)^2} \left( \frac{R_1^2}{c_9} + \frac{R_2^2}{c_7} \right) \\
w_2 &= \frac{PL}{8(R_1 + R_2)} \\
w_3 &= \frac{-P}{12(R_1 + R_2)} \\
w_5 &= \frac{-P(c_7 R_1 - c_9 R_2)^2}{2c_7 c_9 \lambda (c_7 + c_9)(R_1 + R_2)^2} \\
w_6 &= -w_5,
\end{aligned} \tag{12}$$

with

$$\begin{aligned}
\lambda &= \sqrt{\frac{2c_7 c_9 (c_1 + c_3 + c_6)}{(4c_1 c_6 - c_3^2)(c_7 + c_9)}} \\
R_1 &= 2c_1 + c_3 \\
R_2 &= c_3 + 2c_6.
\end{aligned} \tag{13}$$

Interest here will focus on the response at the tip of the cantilever, this being representative of what the beam is doing. In that regard, using  $x=L/2$ , eqs. 11 and 12,

$$\begin{aligned}
w_{\text{tip}} = w\left(\frac{L}{2}\right) &= \frac{PL^3}{48(R_1 + R_2)} + \frac{PL}{4(R_1 + R_2)^2} \left( \frac{R_1^2}{c_9} + \frac{R_2^2}{c_7} \right) \\
&\quad - \frac{P(c_7 R_1 - c_9 R_2)^2}{2c_7 c_9 \lambda (c_7 + c_9)(R_1 + R_2)^2} \left( 1 - e^{-\frac{\lambda L}{2}} \right).
\end{aligned} \tag{14}$$

It has been found that for a very wide range of values of elastic properties, the last term contributes very little. It is thus dropped as it considerably simplifies the algebra.

### Time-Dependent Behavior of the Three-Layer Beam

In the present study, for obtaining an understanding of the overall effects of the time-dependence of the various constituents, and at the same time considering a worst-case scenario, the constituents are modelled as Maxwell fluids. With a Maxwell fluid, for a constant level of applied stress the material strains indefinitely. For a Maxwell fluid the constitutive equation takes the form

$$\sigma + p_1 \dot{\sigma} = q_1 \dot{\varepsilon}, \tag{15}$$

where it is understood that  $\sigma$  can represent a normal or a shear stress and  $\varepsilon$  can represent an extensional or a shear strain. Taking the Laplace transform of both sides results in

$$\mathbb{P}(s)\bar{\sigma}(s) = \mathbb{Q}(s)\bar{\varepsilon}(s), \tag{16}$$

where  $\bar{\sigma}(s)$  and  $\bar{\varepsilon}(s)$  are the Laplace transforms of the stress and strain functions of time, respectively. In the above

$$\begin{aligned} \mathbb{P}(s) &= 1 + p_1 s \\ \mathbb{Q}(s) &= q_1 s \end{aligned} \quad (17)$$

It should be noted that the subscript 1 on  $p_1$  and  $q_1$  in eqs. 15, 17, etc. have nothing to do with the face sheets. The nomenclature  $p_1$  and  $q_1$  are standard for viscoelastic materials. See ref. 2, for example.

If a particular constituent is considered to be time-dependent, then the application of the correspondence principle for the sandwich beam problem at hand calls for replacing, in the formulation for the static elastic response, the elastic modulus of the constituent with the ratio  $\mathbb{Q}(s)/\mathbb{P}(s)$ , i.e.,

$$E \text{ or } G \rightarrow \mathbb{Q}(s)/\mathbb{P}(s) \quad (18)$$

The static load  $P$  is then assumed to be applied at time zero in a stepwise fashion so the load must be replaced with its Laplace transform, i.e.,

$$P \rightarrow P/s \quad (19)$$

The resulting expression is then the Laplace transform of the time-dependent response of the beam. Taking the inverse transform yields the response of the beam as a function of time.

To estimate the long-term effect of time-dependent behavior in the sandwich beams here, it will be assumed that if a constituent exhibits time-dependent behavior, the strain of the constituent will increase by a certain factor in 20 years if subjected to a constant stress. Two levels of time dependency will be studied, one considered worst-case time dependency and the other considered minimal time dependency. The displacement response of the tip for the 20 year period for these two levels of time dependency will then be computed.

As an example, consider the following: With the face sheet material properties being controlled to a large extent by fiber properties, a large degree of time-dependent behavior is unexpected. Hence, it will be assumed that the face sheet material, in the extreme case, exhibits a 10% increase in extensional strain when subjected to a constant stress for 20 years. As a result, for the face sheet material

$$q_1 = 200E_1 \text{ and } p_1 = 200 \text{ years} \quad (20)$$

Using eq. 17, the substitutions indicated by eqs. 18 and 19 can now take place in eq. 14 (with the last term dropped). Performing the inverse transformation results in an expression for the time-dependent tip deflection, namely,

$$\begin{aligned} w_{\text{tip}}(t) &= w_{\text{tip}} + \frac{PL^3}{96} \left( \frac{Aq_1}{BG} \right) \left( 1 - e^{-\frac{B}{G}t} \right) \\ &+ \frac{PL}{16B^2} \left[ \frac{1}{c_9} \left( D^2 - \frac{B^2 H^2}{G^2} \right) - \frac{1}{c_7} \left( \frac{B^2 F^2 q_1^2}{G^2} \right) \right] \left( 1 - e^{-\frac{B}{G}t} \right) \\ &- \frac{PL}{16BG} \left[ \frac{1}{c_9} \left( D - \frac{BH}{G} \right)^2 + \frac{1}{c_7} \left( \frac{B^2 F^2 q_1^2}{G^2} \right) \right] t e^{-\frac{B}{G}t} \end{aligned} \quad (21)$$

where

$$\begin{aligned}
A &= t_1(h^2 + ht_1 + \frac{1}{3}t_1^2) \\
B &= E_2h(\frac{1}{3}h^2) \\
C &= t_1(2h^2 + ht_1) \\
D &= E_2h(\frac{2}{3}h^2) \\
F &= t_1(ht_1 + \frac{2}{3}t_1^2) \\
G &= Aq_1 + Bp_1 \\
H &= Cq_1 + Dp_1
\end{aligned} \tag{22}$$

and  $w_{tip}$  represents the elastic response as given in eq. 14. Consider a sandwich beam with quartz epoxy face sheets in an 8 layer quasi-isotropic lay-up and a honeycomb core. Table 1 illustrates nominal constituent elastic properties. Note that table 1 includes information for the five-layer model to be discussed shortly. For the three-layer model, since the adhesive layer is so thin, the elastic properties of the core are not adjusted to account for lumping the adhesive into the core. Only the thickness of the core is adjusted.

Table 1. Nominal Material Properties and Geometries for Beam Model

| Face Sheets                                                  | Honeycomb Core                                                                                                  | Adhesive                                                                                                        |
|--------------------------------------------------------------|-----------------------------------------------------------------------------------------------------------------|-----------------------------------------------------------------------------------------------------------------|
| $E_1 = 2.5E6$ psi<br>$G_1 = 0.96E6$ psi<br>$t_1 = 0.040$ in. | $E_3 = 1.0E3$ psi<br>$G_3 = 29E3$ psi<br>$h = 0.255$ in.,<br>3-layer model<br>$h = 0.250$ in.,<br>5-layer model | $E_2 = 0.5E6$ psi<br>$G_2 = 0.179E6$ psi<br>$t_2 = 0$ ,<br>3-layer model<br>$t_2 = 0.005$ in.,<br>5-layer model |

From eq. 21 it can be seen that the time dependency is exponential in form but, as shown in fig. 2, over the 20 year period it appears linear. For the quartz-epoxy/honeycomb sandwich, the percent increase in tip deflection, relative to the static elastic deflection at  $t=0$ , is illustrated in fig. 2 for both the worst-case, or maximum, time dependency and the minimal time-dependency of the face-sheet material. Minimal time dependency is defined to be the case when the face-sheet material exhibits only a 1% increase in extensional strain when subjected to a constant tensile stress for 20 years. For the worst case it is seen that the cantilever tip deflection increases by about 8% in 20 years. For the case of minimal time dependency, the tip deflection increases just under 1% in 20 years. These numerical values reflect an almost one-to-one relationship between face-sheet material extensional properties and tip deflection.

As another example, if the core exhibits the behavior of a Maxwell fluid, the appropriate substitutions of eqs. 17, 18, and 19 into eq. 14 leads to an expression for the time-dependent tip deflection as

$$w_{\text{tip}}(t) = w_{\text{tip}} + \frac{PL}{4(R_1 + R_2)^2} \left( \frac{R_1^2}{2h} \right) \frac{t}{q_1} . \quad (23)$$

As can be seen, the tip deflection is linear with time.

Assuming for the case of maximum time dependency that the core strain increases by a factor of 10 in 20 years, and for the case of minimum time dependency that the core strain increases only by a factor of 2 in 20 years, eq. 23 leads to the results shown in fig. 3. It is seen that these cases lead to considerable tip deflection over a 20 year period.

## DEVELOPMENT OF THE FIVE-LAYER MODEL

### Nomenclature and Problem Definition

To explicitly include the adhesive layer, a five-layer model is necessary. The nomenclature and kinematics of the five-layer model are shown in fig. 4. The properties of the face sheets are subscripted with a 1, the properties of the adhesive subscripted with a 2, and the properties of the core subscripted with a 3. Total sandwich thickness is again  $2H$ .

### Equations Governing Elastic Response

The sandwich cross section is again assumed to displace uniformly as-a-whole in the  $x$  direction an amount  $u^o(x)$  and downward an amount  $w^o(x)$ . The cross sections of the face sheet layers are assumed to rotate independently of the cross sections of the adhesive layers, and the core has its own cross-section rotation. The displacement field is thus given by

$$u(x,z) = \begin{array}{ll} u^o(x) + h\alpha(x) + t_2\beta(x) - (z + h + t_2)\gamma(x) & -H \leq z \leq -H + t_1 \\ u^o(x) + h\alpha(x) - (z + h)\beta(x) & -H + t_1 \leq z \leq -h \\ u^o(x) - z\alpha(x) & -h \leq z \leq h \\ u^o(x) - h\alpha(x) - (z - h)\beta(x) & h \leq z \leq H - t_1 \\ u^o(x) - h\alpha(x) - t_2\beta(x) - (z - h - t_2)\gamma(x) & H - t_1 \leq z \leq H \end{array} \quad (24)$$

$$w(x,z) = w^o(x) .$$

There are five kinematic quantities to be determined with the five-layer model. Dropping the superscript  $o$ , the strains are given by

$$\begin{aligned}
\varepsilon_x = & \begin{aligned} & \frac{du}{dx} + h \frac{d\alpha}{dx} + t_2 \frac{d\beta}{dx} - (z + h + t_2) \frac{d\gamma}{dx} & -H \leq z \leq -H + t_1 \\ & \frac{du}{dx} + h \frac{d\alpha}{dx} - (z + h) \frac{d\beta}{dx} & -H + t_1 \leq z \leq -h \\ & \frac{du}{dx} - z \frac{d\alpha}{dx} & -h \leq z \leq h \\ & \frac{du}{dx} - h \frac{d\alpha}{dx} - (z - h) \frac{d\beta}{dx} & h \leq z \leq H - t_1 \\ & \frac{du}{dx} - h \frac{d\alpha}{dx} - t_2 \frac{d\beta}{dx} - (z - h - t_2) \frac{d\gamma}{dx} & H - t_1 \leq z \leq H \end{aligned} \\
\gamma_{xz} = & \begin{aligned} & -\gamma + \frac{dw}{dx} & -H \leq z \leq -H + t_1 \\ & -\beta + \frac{dw}{dx} & -H + t_1 \leq z \leq -h \\ & -\alpha + \frac{dw}{dx} & -h \leq z \leq h \\ & -\beta + \frac{dw}{dx} & h \leq z \leq H - t_1 \\ & -\gamma + \frac{dw}{dx} & H - t_1 \leq z \leq H, \end{aligned} \tag{25}
\end{aligned}$$

Substituting these strains into the total potential energy, eq. 4, leads to the following differential equations for the kinematic variables:

$$\begin{aligned}
2c_0 \left( \frac{d^2 u}{dx^2} \right) &= 0 \\
2c_1 \left( \frac{d^2 \alpha}{dx^2} \right) + c_2 \left( \frac{d^2 \beta}{dx^2} \right) + c_3 \left( \frac{d^2 \gamma}{dx^2} \right) + c_9 \left( \frac{dw}{dx} - \alpha \right) &= 0 \\
c_2 \left( \frac{d^2 \alpha}{dx^2} \right) + 2c_4 \left( \frac{d^2 \beta}{dx^2} \right) + c_5 \left( \frac{d^2 \gamma}{dx^2} \right) + c_8 \left( \frac{dw}{dx} - \beta \right) &= 0 \\
c_3 \left( \frac{d^2 \alpha}{dx^2} \right) + c_5 \left( \frac{d^2 \beta}{dx^2} \right) + 2c_6 \left( \frac{d^2 \gamma}{dx^2} \right) + c_7 \left( \frac{dw}{dx} - \gamma \right) &= 0 \\
c_7 \left( \frac{d\gamma}{dx} - \frac{d^2 w}{dx^2} \right) + c_8 \left( \frac{d\beta}{dx} - \frac{d^2 w}{dx^2} \right) + c_9 \left( \frac{d\alpha}{dx} - \frac{d^2 w}{dx^2} \right) &= 0
\end{aligned} \tag{26}$$

where the constants  $c_i$  are given by



$$\begin{aligned}
c_0 &= E_1 t_1 + E_2 t_2 + E_3 h \\
c_1 &= h^2(E_1 t_1 + E_2 t_2 + \frac{1}{3} E_3 h) \\
c_2 &= h t_2(2E_1 t_1 + E_2 t_2) \\
c_3 &= h t_1(E_1 t_1) \\
c_4 &= t_2^2(E_1 t_1 + \frac{1}{3} E_2 t_2) \\
c_5 &= t_1 t_2(E_1 t_1) \\
c_6 &= t_1^2(\frac{1}{3} E_1 t_1) \\
c_7 &= 2G_1 t_1 \\
c_8 &= 2G_2 t_2 \\
c_9 &= 2G_3 h .
\end{aligned} \tag{27}$$

The relevant boundary conditions are

$$\begin{aligned}
&\text{at } x = 0: & \text{at } x = \frac{L}{2}: \\
u = 0 & & 2c_0 \left( \frac{du}{dx} \right) = 0 \\
\alpha = 0 & & 2c_1 \left( \frac{d\alpha}{dx} \right) + c_2 \left( \frac{d\beta}{dx} \right) + c_3 \left( \frac{d\gamma}{dx} \right) = 0 \\
\beta = 0 & & c_2 \left( \frac{d\alpha}{dx} \right) + 2c_4 \left( \frac{d\beta}{dx} \right) + c_5 \left( \frac{d\gamma}{dx} \right) = 0 \\
\gamma = 0 & & c_3 \left( \frac{d\alpha}{dx} \right) + c_5 \left( \frac{d\beta}{dx} \right) + 2c_6 \left( \frac{d\gamma}{dx} \right) = 0 \\
w = 0 & & (c_7 + c_8 + c_9) \left( \frac{dw}{dx} \right) - c_7 \gamma - c_8 \beta - c_9 \alpha = \frac{P}{2} .
\end{aligned} \tag{28}$$

As before, the equation governing  $u^o(x)$  decouples from the other four and it can be disregarded at this time. Solution of the equations for  $\alpha(x)$ ,  $\beta(x)$ ,  $\gamma(x)$ , and  $w(x)$  follows the procedure for the three-layer beam. The algebra, however, is considerably more involved. As with the three-layer beam, approximations are used to simplify the algebra, particularly for application of the correspondence principle. Through these approximations, the expression for the tip deflection of the five-layer beam takes the form

$$\begin{aligned}
w_{\text{tip}} = w\left(\frac{L}{2}\right) &= \frac{PL^3}{48(R_1 + R_2 + R_3)} \\
&+ \frac{PL}{4(R_1 + R_2 + R_3)^2} \left( \frac{R_1^2}{c_9} + \frac{R_2^2}{c_8} + \frac{R_3^2}{c_7} \right) .
\end{aligned} \tag{29}$$

where

$$\begin{aligned}
R_1 &= 2c_1 + c_2 + c_3 \\
R_2 &= c_2 + 2c_4 + c_5 \\
R_3 &= c_3 + c_5 + 2c_6 .
\end{aligned} \tag{30}$$

This equation is the analog of eq. 14 for the three-layer beam.

## Time-Dependent Response of the Five-Layer Beam

Numerical studies with the five-layer beam indicate that little new information, relative to what can be learned with the three-layer model, is obtained by using the five-layer model to study the influence of time-dependent face sheet and core properties. However, since the five-layer model includes the additional feature of the adhesive layer, it is of value to discuss the influence of time-dependent adhesive properties. Assuming the adhesive shear properties behave as a Maxwell fluid, using the expression for the tip deflection, eq. 29, performing the steps given by eqs. 18 and 19, and taking the inverse transformation, the tip deflection as a function of time for a viscoelastic adhesive layer is

$$w_{\text{tip}}(t) = w_{\text{tip}} + \frac{PL}{4(R_1 + R_2 + R_3)^2} \left( \frac{R_2^2}{2t_2} \right) \left( \frac{t}{q_1} \right). \quad (31)$$

This expression also depends on time in a linear manner. Using the material properties as given in table 1, the time-histories of the percent increase in tip deflection for the cases of the adhesive shear strain doubling and increasing ten-fold in 20 years are shown in fig. 5. For the worst-case condition, the tip deflection changes by less than 0.5% in 20 years.

## CONCLUSIONS

Presented has been the development of models to be used in evaluating the influence of time-dependent face sheet, core, and adhesive constitutive properties on the overall deformations of sandwich beams. The study has its origins in the need to understand the time-dependent deformations of orbiting precision segmented reflectors. Beams may be considered an oversimplification of the structural characteristics of segmented reflectors. However, the basic characteristics of sandwich construction are retained in the beam models, and beams could serve as a screening tools as effectively, and certainly as economically, as plate or shell-like models. Here efforts have been made to involve the important material properties explicitly so parametric studies can easily be made. Some approximations were necessary, but these have been justified and in no way do they compromise the results obtained.

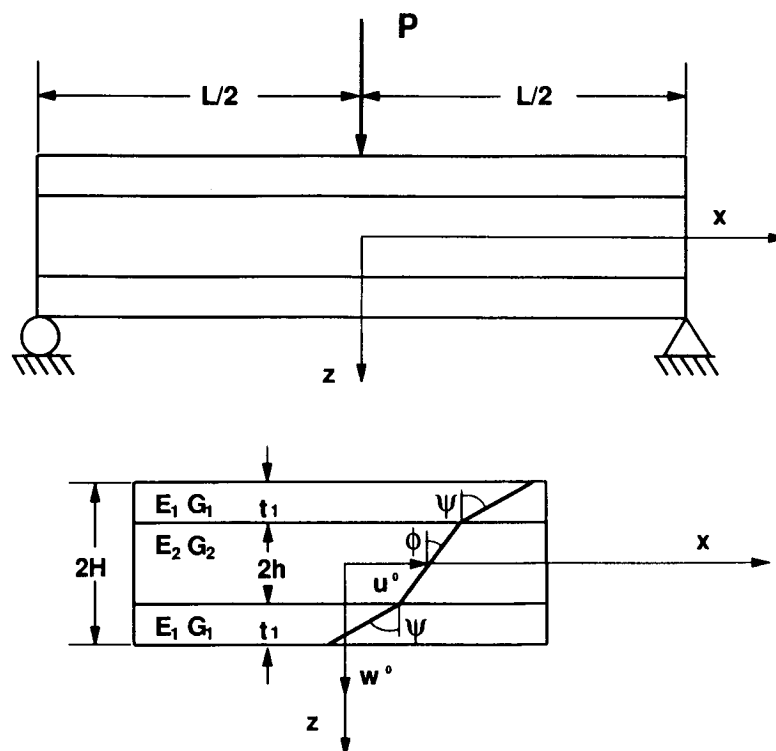
Several recommendations are in order. First, extending the analysis to include thermal effects, such as would occur in the presence of a slight through-the-thickness temperature gradient, would be worthwhile. Such a gradient would cause unwanted curvature in a beam, and over time, the curvature may change. Second, it would be useful to extend the analysis to include the two-dimensional aspects of the reflector, namely its plate-like geometry. If the change of focal length with time, for example, of a reflector is to be determined, such an analysis is necessary.

## ACKNOWLEDGMENT

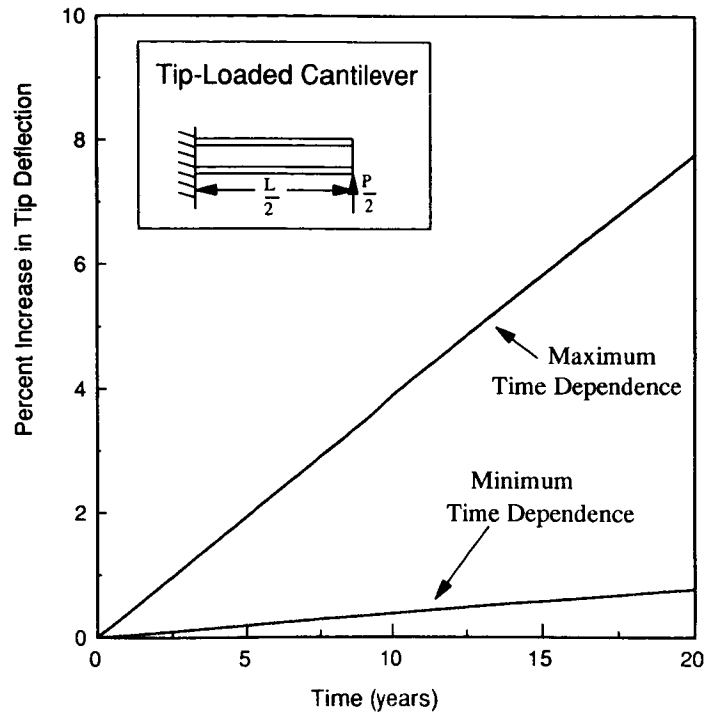
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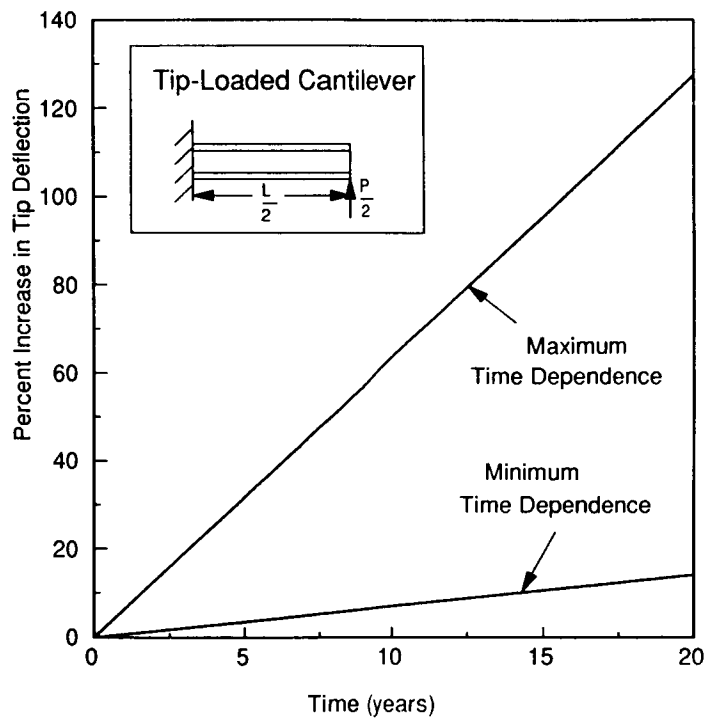
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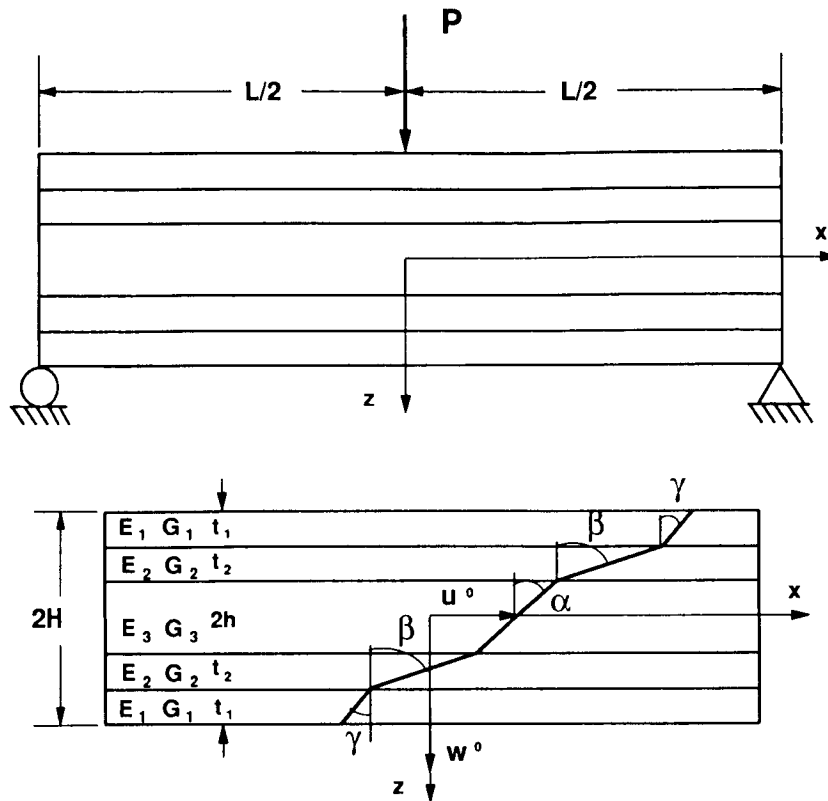
**Fig. 1 - Geometry, nomenclature, loading, and assumed displacement field for the three-layer model.**



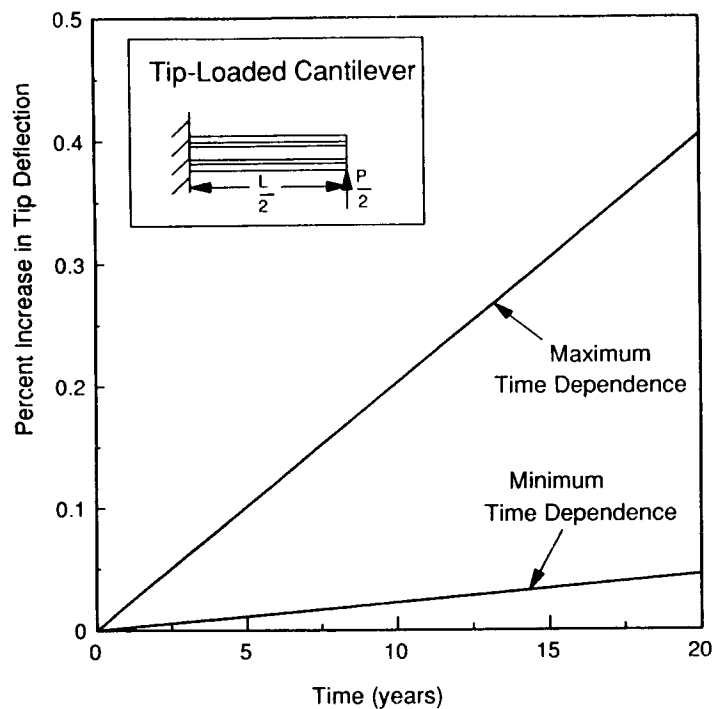
**Fig. 2 - Percent increase in tip deflection of three-layer beam due to time-dependent face-sheet extensional properties.**



**Fig. 3 - Percent increase in tip deflection of three-layer beam due to time-dependent core shear properties.**



**Fig. 4 - Geometry, nomenclature, loading, and assumed displacement field for the five-layer model.**



**Fig. 5 - Percent increase in tip deflection of five-layer beam due to time-dependent adhesive layer shear properties.**

# COMPOSITE FLEXIBLE INSULATION FOR THERMAL PROTECTION OF SPACE VEHICLES

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## SUMMARY

A composite flexible blanket insulation (CFBI) system considered for use as a thermal protection system for space vehicles is described. This flexible composite insulation system consists of an outer layer of silicon carbide fabric, followed by alumina mat insulation, and alternating layers of aluminized polyimide film and aluminoborosilicate scrim fabric. A potential application of this composite insulation would be as a thermal protection system for the aerobrake of the Aeroassist Space Transfer Vehicle (ASTV). It would also apply to other space vehicles subject to high convective and radiative heating during atmospheric entry. The thermal performance of this composite insulation as exposed to a simulated atmospheric entry environment in a plasma arc test facility is described. Other thermophysical properties which affect the thermal response of this system are also described. Analytical modeling describing the thermal performance of this composite insulation is included. It shows that this composite insulation is effective as a thermal protection system at total heating rates up to  $30.6 \text{ W/cm}^2$ .

## INTRODUCTION

Composite flexible blanket insulation (CFBI) is multilayer insulation consisting of ceramic fabrics, insulation, and reflective foils that are separated by ceramic scrim cloths. A potential application for such multilayer insulations would be as a thermal protection system for the aerobrake of the Aeroassist Space Transfer Vehicle (ASTV). Depending on the exact location of the insulation on the aerobrake, this insulation would be exposed to heating rates up to approximately  $31 \text{ W/cm}^2$  (ref. 1).

The maximum heating rate will be reached at approximately 100 seconds into the heating pulse. The multilayer insulation exposed to these heating rates would reach maximum backface temperature at approximately 200 seconds at which time the pressure is essentially that of space. Multilayer insulation is intended for use in the space vacuum where gas conductivity between the foils is negligible and the overall effective thermal conductivity is very small. This appropriately designed multilayer insulation could operate efficiently within the heating and pressure environment of the ASTV, providing a weight saving compared to other types of insulation. In order to demonstrate the effectiveness of this insulation in a typical ASTV flight environment, it is planned to install and fly this type of insulation on the aerobrake of a precursor of the ASTV called Aeroassist Flight Experiment (AFE) (ref. 2).

## DESCRIPTION OF MATERIALS

A description of a typical composite flexible blanket insulation is given in figure 1. The composite insulation consists of an outer layer of silicon carbide fabric, followed by an alumina insulation and alternating layers of aluminized polyimide films. It also includes aluminoborosilicate scrim cloths and a bottom layer of aluminoborosilicate fabric. The entire insulation is sewn with a silicon carbide thread.

Some of the properties of the components in this composite flexible blanket insulation are shown in table 1. The overall thermal performance of this insulation is greatly dependent on the thermo-physical properties of the components and the physical arrangement of the components to form the insulation system. Heat transfer in multilayer insulations usually occurs through conduction, convection, and radiation. For space or vacuum applications, the convection mechanism may be neglected as the gas phase is at a greatly reduced pressure. The heat transport processes which are to be considered are conduction through the solid phase of the insulation and radiation. The radiation becomes the dominant mechanism as temperature increases, whereas conduction determines the lower limit of thermal conductivity at lower temperatures. The multilayer insulations have a very small separation such as scrim cloth between the foils. An approximation of the total heat transfer in the full configuration may be estimated by treating the individual components independently. The total heat flux,  $\dot{q}$ , through this type of insulation may be expressed as follows:

$$\dot{q}(\text{total}) = \dot{q}(\text{radiation}) + \dot{q}(\text{gas conduction}) + \dot{q}(\text{solid conduction}) \quad (1)$$

The design of the composite insulation shown in figure 1 was based primarily on the theory of equation (2) that describes the heat transfer in multilayer insulations. This equation (ref. 1) is as follows:

$$K_e = BK_s D_f n + \frac{(r)^2 G(T_F^2 + T_B^2)(T_F + T_B)t}{(a + 2s)(t/2) + (N - 1)[(2/E) - 1]} + \frac{(L)}{(L + I)} (K_g) \quad (2)$$

where

|       |                                              |
|-------|----------------------------------------------|
| $K_e$ | effective thermal conductivity of insulation |
| $B$   | solid fraction in insulation                 |
| $K_s$ | thermal conductivity of solids               |
| $D_f$ | fiber diameter                               |
| $n$   | exponent                                     |
| $r$   | index of refraction                          |
| $G$   | Stefan-Boltzman constant                     |
| $T_F$ | temperature on front side                    |
| $T_B$ | temperature on back side                     |
| $t$   | thickness of insulation                      |
| $L$   | cell length in insulation                    |
| $a$   | absorption cross section                     |
| $s$   | scattering cross section                     |
| $N$   | number of metal foils                        |
| $E$   | emittance                                    |
| $I$   | mean free path of gas                        |
| $K_g$ | thermal conductivity of gas                  |

Equation (2) was used as a guide for the selection of materials and geometry of the composite insulation. It shows the importance of some of the properties of the components and geometry in the multilayer insulations. For example, to achieve lower thermal conductivity in the entire system, a larger number of foils having a low emittance at elevated temperatures is desired. The foils should be placed in the composite where  $T_F$  and  $T_B$  are relatively small. A small fiber diameter is desired in the insulation as well as low thermal conductivity of the components. Equation (2) shows the influence of the optical properties of both the insulation and foil materials and the number of foils. It also shows the physical and thermal properties of the components (such as density and thickness of insulation), fiber diameter, and the boundary temperatures on the heat transfer properties of the multilayer assembly.



Using these parameters and previous tests (refs. 2 and 3) as guides, the multilayer insulation shown in figure 1 was fabricated and its heat transfer properties evaluated. The reflective shield used in this insulation is a polyimide film with a chemical vapor-deposited aluminum film approximately 800 micrometers thick. As shown in the above equation, a low emittance is desired in the reflective shield. The emittance of this film up to 673 K is approximately 0.05 (ref. 4). The low emittance and high reflectance lead to the effectiveness of these films as a radiation shield in the heating environment of the AFE vehicle. The expected peak heating rate at this environment is approximately  $30.6 \text{ W/cm}^2$  consisting of approximately  $4.7 \text{ W/cm}^2$  radiative heating and  $25.9 \text{ W/cm}^2$  convective (ref. 5). The total heating and pressure rate to which these insulations will be exposed in the space environment is shown in figure 2. As shown, the surface of the insulation will reach the maximum heating rate at approximately 100 seconds. It will be maintained at this rate for approximately 50-60 seconds, after which time both the heating rate and pressure decrease. The "Nominal" and "3-sigma" surface temperature profiles of the AFE Aerobrake at the CFBI location is shown in figure 3. This temperature is the surface temperature on the fibrous refractory composite insulation/ reaction cured glass (FRCI/RCG) at this location. The purpose of the composite flexible insulation is to provide thermal protection to the aluminum substructure when exposed to the heating and pressure environment shown in figure 2. As a guideline, the maximum acceptable temperature limit for the aluminum substructure has been defined as 445.5 K (ref. 1).

The apparent thermal conductivity of the composite flexible insulation shown in figure 1 was determined using the procedure described in reference 6. The apparent thermal conductivity as a function of temperature at three different pressures is shown in figure 4. The thermal conductivity of the alumina insulation used in the CFBI is also shown in the same figure. The thermal conductivity of the CFBI is lower than that of the alumina insulation at lower temperatures but slightly higher at higher temperatures. This could be attributed to the silicon carbide fabric and thread. In addition, the heating in this test method is entirely convective.

## TEST RESULTS

One of the objectives of the AFE is to determine the thermal response of this type of insulation under the AFE environment. Another objective is to compare with state-of-the-art rigid thermal insulations such as the fibrous refractory composite insulation (FRCI) coated with a reaction cured glass (RCG) coating described previously (refs. 7-9). FRCI is considered as the baseline tile for the heating region where the composite flexible insulation will be located. The purpose of the tests reported here was to attempt to simulate, as closely as possible, the temperature and heating rate conditions of the AFE trajectory in a test facility. Another purpose is to determine the thermal response of these composite insulations under this environment. The test facility utilized was the NASA Ames 20-MW Plasma Arc Facility described in reference 9. Three radiation equilibrium temperature conditions were used to evaluate the insulations. These temperatures were based on three different heating rates resulting from three different trajectories. These heating rates are  $30.6 \text{ W/cm}^2$  for a "nominal" condition,  $35.2 \text{ W/cm}^2$  for a "3-sigma" condition and  $39.7 \text{ W/cm}^2$  for a "peak heating rate" condition.

These conditions represent the surface temperatures resulting from the "Nominal" and "3-sigma" trajectories of a 1857 kg AFE vehicle and the temperature resulting from the "Peak Heating Rate" of

a 2038 kg AFE vehicle (ref. 10). The model used to establish the test conditions consisted of an FRCI tile approximately  $9\text{ cm} \times 2.5\text{ cm}$  thick installed in the center of another FRCI holder approximately  $16\text{ cm}$  in diameter  $\times 5\text{ cm}$  thick. These test conditions produced the FRCI tile surface temperature profiles shown in figure 5. The center rigid model was subsequently replaced with the composite flexible insulation of the same dimensions in the larger holder. The purpose of the larger holder was to reduce any side heating effects on the insulation during testing. The flexible insulation was bonded on the bottom side with a silicone adhesive to an aluminum plate  $0.08\text{ cm}$  thick.

The rigid calibration model was instrumented with three thermocouples located, within the RCG coating, on the surface of the tile. The composite flexible insulation test model shown in figure 6, was instrumented with a thermocouple encased in alumina tubing and bonded, with alumina adhesive, on the underside of the top fabric shown in figure 1 and described in table 1. A second thermocouple was installed within the silicone adhesive, between the bottom fabric of the CFBI and the aluminum plate. Both thermocouples were located on the geometric center of the CFBI model.

The surface temperature profiles for the FRCI/RCG calibration model are shown in figure 6. The temperature profiles achieved were fairly close to the targeted temperature profiles shown in figure 3 except for the "heat-up" rate, which is very rapid in the test. The models were inserted in the plasma arc stream for 120 seconds, which is slightly longer than the time of the peak temperature of the AFE. The pressure in the arc jet test chamber was maintained at  $0.27\text{ Pa}$  ( $0.004\text{ psi}$ ) during the test model exposure in the arc stream. Subsequently, it was increased in three (3) stages for 440 seconds during the "cool down" of the model to approximate the pressure profile of the AFE space vehicle. The equivalent FRCI/RCG heating rates achieved for the three test conditions were  $30\text{ W/cm}^2$ ,  $35\text{ W/cm}^2$  and  $39\text{ W/cm}^2$  respectively.

The thermal response of the composite flexible blanket insulation at the  $30\text{ W/cm}^2$  test condition is shown in figure 7. The surface temperature is the temperature recorded from a Type R thermocouple encased in alumina tubing and bonded on the bottom side of the silicon carbide fabric. The backface temperature is the average temperature recorded by two Type K thermocouples placed within the silicone adhesive. The surface temperature is higher than the FRCI/RCG surface temperature shown in figure 6 at equivalent test conditions. This is attributed to the lower emissivity of the top fabric and other factors such as fabric design and texture. No visual degradation was observed in the fabric. The maximum surface temperature attained was  $1850\text{ K}$ . The maximum backface temperature attained was  $450\text{ K}$  at 230 seconds. This is within the design limit for the aluminum structure of the AFE vehicle.

The thermal response of the same insulation at the  $35\text{ W/cm}^2$  test condition is shown in figure 8. This test condition represents the "3-sigma" trajectory of the 1857-kg AFE vehicle. The maximum surface temperature was  $1900\text{ K}$  and the maximum backface temperature was  $500\text{ K}$  at 230 seconds after the model insertion in the arc stream. Again, the surface temperature is considerably higher than the FRCI/RCG surface temperature. No visual degradation of the surface fabric was observed except for some slight discoloration. The back aluminum plate in the test insulation was  $0.08\text{ cm}$  thick. The aluminum skin on the aerobrake is the same thickness, but it is supported by aluminum stringers resulting in an effective thickness of  $0.28\text{ cm}$  thick. A thicker backplate in the test model could have resulted in a lower backface temperature since it would act as a heat sink.

The thermal response of the insulation at the 39 W/cm<sup>2</sup> condition is shown in figure 9. The surface temperature reached 1750 K and the backface 580 K at 200 seconds. The surface fabric was removed during testing at some locations possibly due to oxidation and tensile failure of the yarn. This test condition exceeds any of the 1857-kg vehicle trajectory conditions and is not likely to be encountered in flight.

## ANALYTICAL MODELING

As stated previously, the composite flexible blanket insulation was instrumented on the surface with a thermocouple probe. The probe consisted of a thermocouple inserted in an alumina tubing bonded to the silicon carbide fabric. The purpose of the tubing is to protect the thermocouple during the thermal exposure and to assure full contact with the surface of the fabric.

Simulation models were developed using the system improved numerical differencing analyzer (SINDA) program (ref. 11) to analyze the thermal responses of both the CFBI insulation and the thermocouple probe. The specific objectives were to predict the in-depth temperatures of the CFBI when exposed to the nominal AFE trajectory and to correlate the temperatures measured by the thermocouple probe to those of the SiC surface.

The models assume an imposed heat flux history at the front surface. Heat is then reradiated to the surrounding blanket as well as conducted through the blanket. The remainder of the heat produces changes in material temperatures. The emissivity of SiC at high temperature has not been well characterized. For these calculations, it was assumed to vary between 0.86 at 311 K and 0.48 at 1839 K (ref. 12). The conductivity of CFBI with silica batting used in the analysis is shown in figure 10, as a function of both temperature and pressure. The models linearly interpolate or extrapolate the thermal properties as required. The backface of the aluminum was taken to be adiabatic. It should be noted that the model did not include the multifoil assembly due to its complexity, which results in conservative calculation of the backface temperature.

A one-dimensional SINDA model was used to determine the in-depth temperature response of CFBI. Figure 11 is a schematic of the model components, including the SiC fabric, alumina batting, RTV and aluminum skin. Figure 12 shows the surface, in-depth, and backface temperatures calculated for the nominal AFE aerothermal environment shown in figure 2. The calculated backface temperature reaches a maximum of 440 K, which is within the aluminum temperature limit of 450 K. The backface of the test sample exposed to the 30 W/cm<sup>2</sup> test condition reached a maximum backface temperature of 450 K, indicating a good correlation of the experimental and calculated results.

Because of the thermal inertia and insulation effects of the alumina probe, the temperature measured by the thermocouple lags the surface temperature. This phenomena was evident from experimental arc-jet results. A three-dimensional SINDA model was developed and used to study the thermal response of the thermocouple probe and to correlate its temperature to the SiC fabric temperature. Figure 13 details the geometry and components of the model.

Comparisons were made between the calculated thermocouple temperature against the experimental data in order to validate the three-dimensional model. However, the actual heat fluxes to which the test sample was subjected were not directly available. Instead, the heat fluxes were calculated by using the RCG temperatures measured from the calibration test, since the thermal physical properties of the RCG and FRCI are well established (ref. 8). Using a constant RCG emissivity of 0.85 and assuming the measured RCG temperatures were at radiation equilibrium, the heat fluxes were then calculated.

After the heat fluxes had been established for the test conditions, they were then imposed onto the three-dimensional model. The model simulates the thermal response of the thermocouple probe embedded in the CFBI. Figure 14 compares the calculated thermocouple probe temperatures to the empirical values. The results indicated that the model predictions correspond well with the measured values. However, the model predictions still lag behind the actual measurements. The discrepancy was probably due to the difference between the conductivities used in the analysis and those of the actual insulation material. Another possible cause is the added thermal mass of the adhesive used to secure the probe. Further refinement of the model will be carried out. Figure 14 also shows the calculated top surface temperatures at a location away from the thermal effect of the probe. As expected, the SiC temperatures respond significantly faster than the probe temperatures.

## CONCLUSIONS

The thermal properties of a lightweight composite flexible blanket insulation suitable for the Aeroassist Space Transfer Vehicle environment were reviewed. This multilayer type insulation provides thermal protection to an aluminum structure when exposed to a heating environment similar to that of the aerobrake of the Aeroassist Flight Experiment. No visual failure of the insulations is observed when the insulations are exposed to a plasma arc test condition which produces a surface temperature of 1645 K on an FRCI/RCG rigid insulation. The surface temperature of the insulation is higher and this is attributed to the lower emissivity of the silicon carbide fabric. The ability of the thermal analysis models to predict the temperature responses of these insulations was demonstrated. Further refinements to these models are being carried out. They will be utilized in the future to aid analyses of the temperatures obtained during the actual AFE flight.

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**Table 1. Typical properties of components in composite flexible blanket insulation.**

| Component                                                                                                                                                         | Thickness,<br>cm                | Density,<br>g/cm <sup>3</sup> | Area<br>density,<br>g/m <sup>2</sup> | Specific<br>heat,<br>kJ/kg K | Thermal<br>conductivity,<br>W/m K                  |
|-------------------------------------------------------------------------------------------------------------------------------------------------------------------|---------------------------------|-------------------------------|--------------------------------------|------------------------------|----------------------------------------------------|
| Silicon carbide fabric<br>5 harness satin weave<br>Yarn count<br>1260 warp × 670<br>Fill/m, $1.5 \times 10^3$<br>Filaments/m <sup>2</sup><br>Fiber diameter, 9 μm | 0.065                           |                               | 570                                  |                              | 0.664 along fiber<br>axis at 300 K at<br>101.3 kPa |
| Alumina mat (95%<br>Al <sub>2</sub> O <sub>3</sub> 5% SiO <sub>2</sub> )<br>Fiber diameter, 3 μm                                                                  | 2.305                           | 0.096                         | 2213                                 | 0.336 at<br>1200 K           | 0.15 at 1200 K at<br>101.3 kPa                     |
| Aluminoborosilicate fab-<br>ric (62% Al <sub>2</sub> O <sub>3</sub> , 24%<br>SiO <sub>2</sub> , 14% B <sub>2</sub> O <sub>3</sub> )<br>Fiber diameter, 3 μm       | 1 ply × 0.33<br>9 plies × 0.010 |                               | 258<br>9 plies × 34                  |                              |                                                    |
| Aluminized polyimide<br>film                                                                                                                                      | 10 plies ×<br>0.007             |                               | 10 plies ×<br>11                     |                              |                                                    |
| Aluminoborosilicate fab-<br>ric (62% Al <sub>2</sub> O <sub>3</sub> , 24%<br>SiO <sub>2</sub> , 14% B <sub>2</sub> O <sub>3</sub> )<br>Fiber diameter, 3 μm       | 0.109                           |                               | 791                                  |                              |                                                    |
| Silicon carbide thread                                                                                                                                            | 0.036                           |                               | 300                                  |                              |                                                    |
| Composite flexible blan-<br>ket insulation assembly                                                                                                               | 2.61                            | 0.174                         | 4548                                 |                              | 0.11 at 1250 K at<br>101.3 kPa                     |

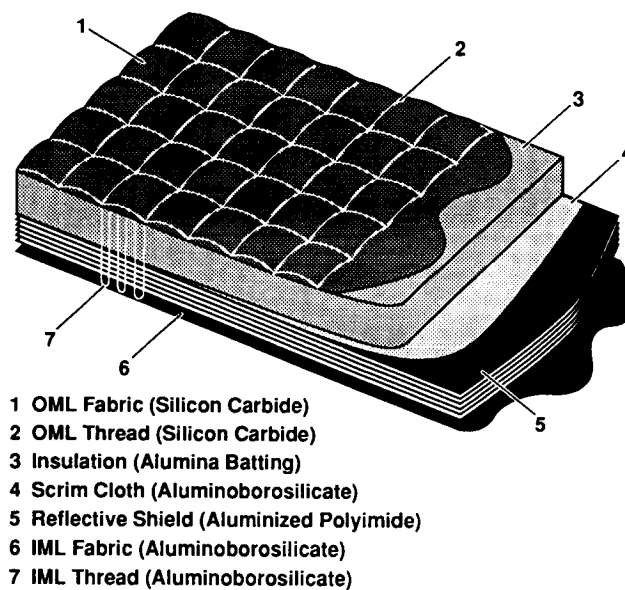


Figure 1. Composite flexible blanket insulation (CFBI).

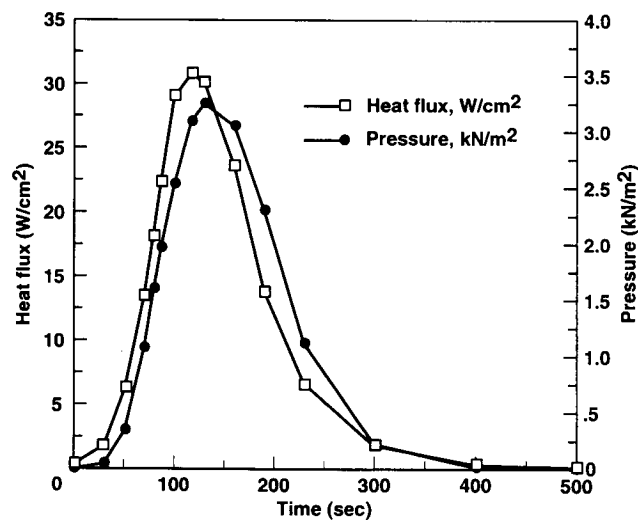


Figure 2. Heating and pressure profiles of the AFE aerobrake at the composite flexible blanket insulation location.

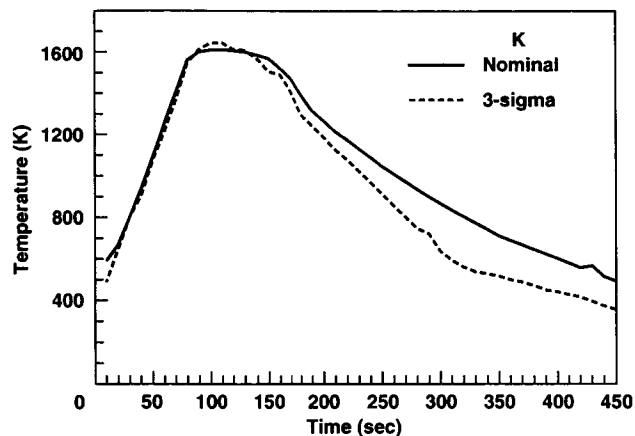


Figure 3. Nominal and three-sigma surface temperature profiles of the AFE aerobrake at the composite flexible blanket insulation location.

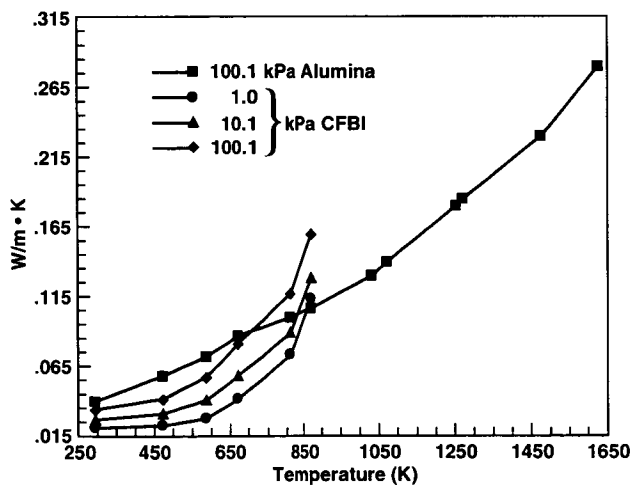


Figure 4. Apparent thermal conductivity of composite flexible blanket insulation.



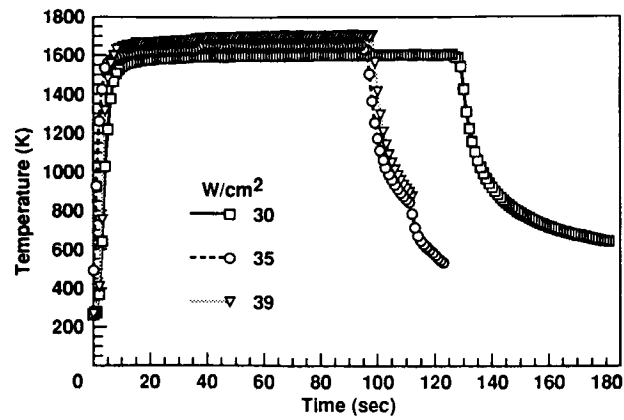


Figure 5. Surface temperatures of fibrous refractory composite insulation/reaction cured glass in the 20 mW Plasma Arc Test Facility.

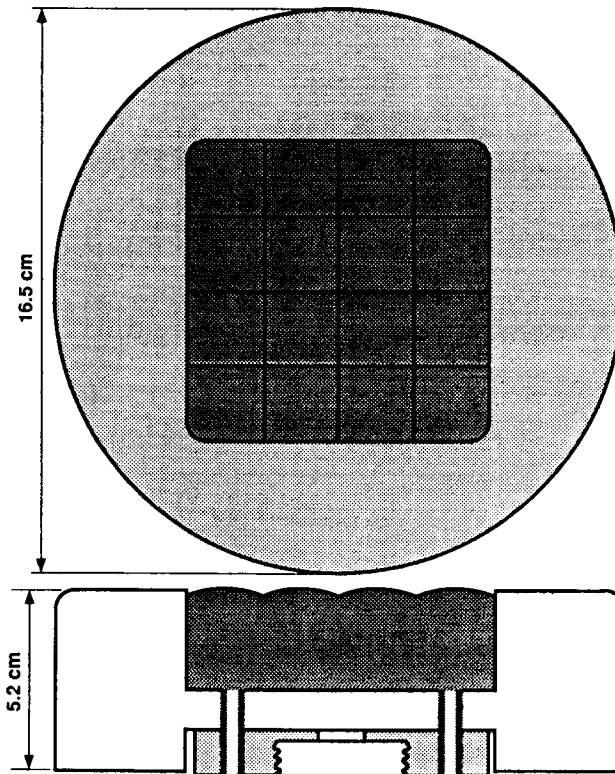


Figure 6. Arc-jet test model for flexible insulation.

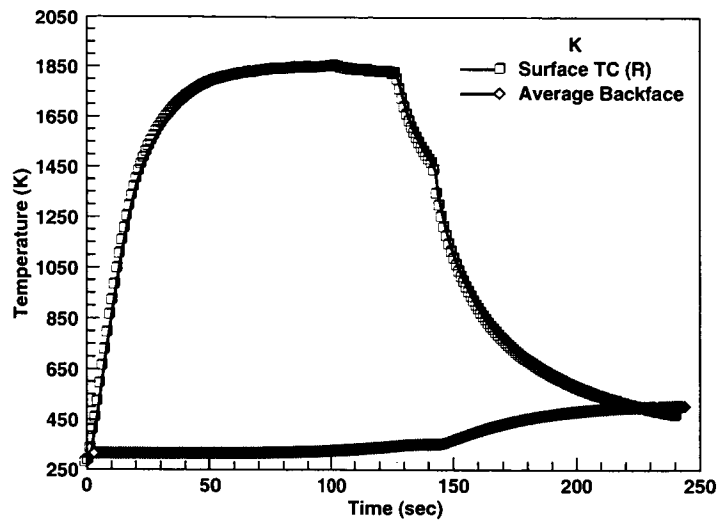


Figure 7. Thermal response of composite flexible blanket insulation at 30 W/cm<sup>2</sup> test condition in the 20 mW Plasma Arc Test Facility. TC – thermocouple.

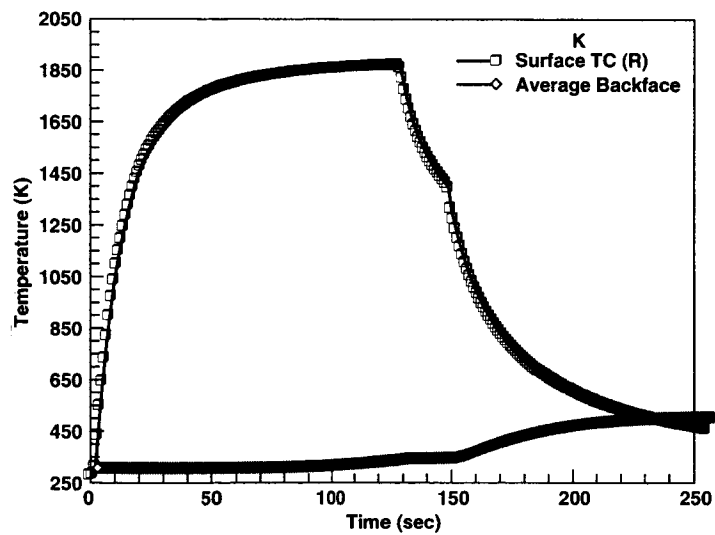


Figure 8. Thermal response of composite flexible blanket insulation at 35 W/cm<sup>2</sup> test condition in the 20 mW Plasma Arc Test Facility.

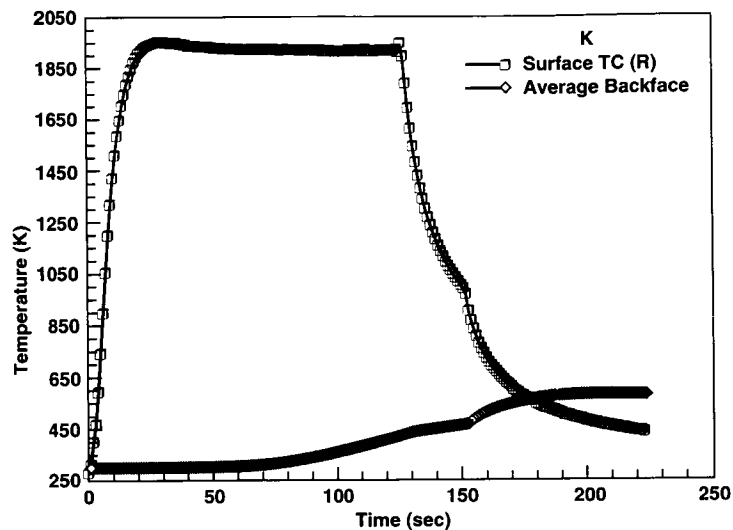


Figure 9. Thermal response of composite flexible blanket insulation at  $39 \text{ W/cm}^2$  test condition in the 20 mW Plasma Arc Test Facility.

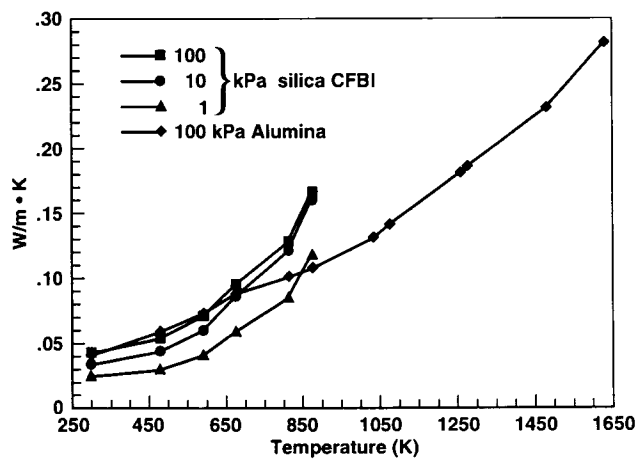


Figure 10. Apparent thermal conductivity of composite flexible blanket insulation (CFBI) with silica batting.

# **1-D THERMAL MATH MODEL** used for CFBI In-Depth Thermal Conditions

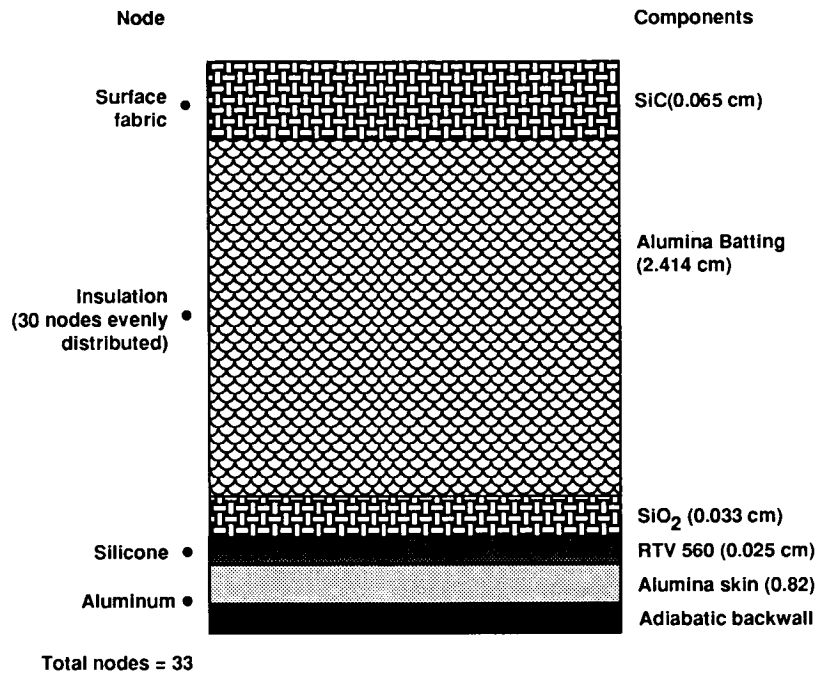


Figure 11. Heat transfer model for composite flexible blanket insulation. RTV – room temperature vulcanizing.

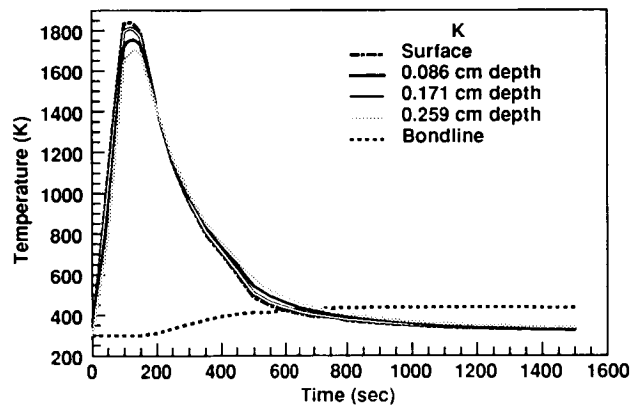
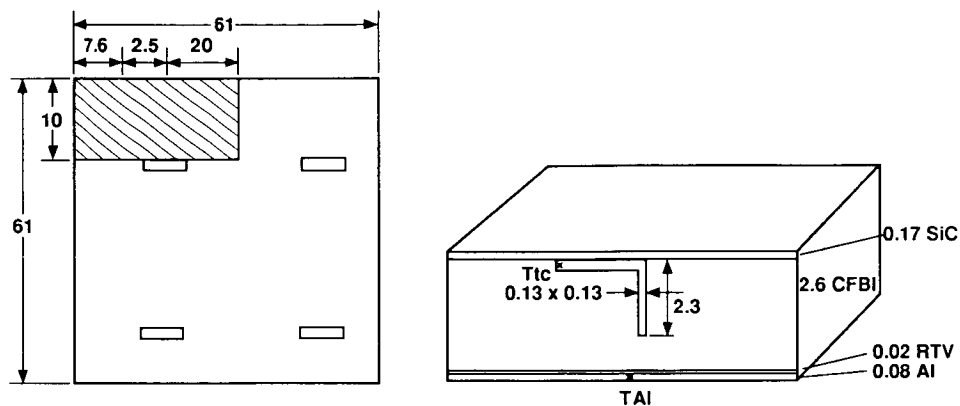


Figure 12. Calculated thermal response of composite flexible blanket insulation during nominal aeroassist flight experiment trajectory.



Note: All dimensions in centimeters

Figure 13. Model geometry of composite flexible blanket insulation with thermocouple probe.

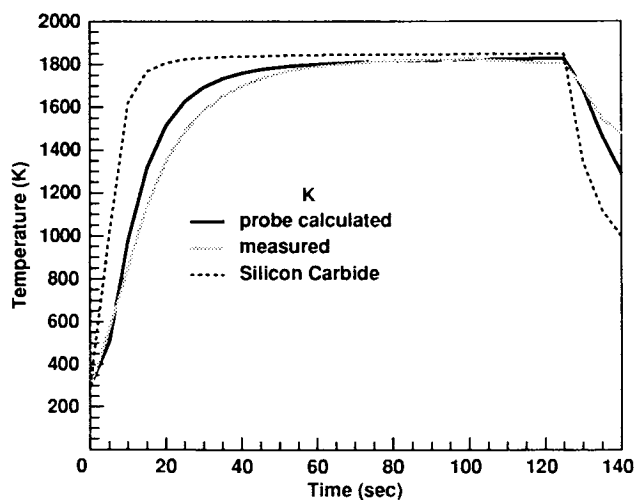


Figure 14. Comparison of calculated and measured surface temperature of composite flexible blanket insulation at 1589 K test condition.

## Technical Report Documentation Page

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